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31 CHRISTOPHER ST., LONDON, E. C.

E. J. P. Benn, Director and Publisher
Associate Inst. N. A.

Edited by H. H. Brown, A. M. Inst. N. A.
Member Soc. N. A. and M. E.

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The Construction of Steel River Barges

General Types of River Barges for Transportation of
Bulk and Package Freight—Details of Construction

In the development of water-borne traffic on the inland waterways of the United States, three essential requirements must be met: First, there must be a sufficient stage of water; second, there must be ample terminal facilities, and, third, there must be suitable craft. When the foregoing essentials are supplied, freight in satisfactory volume will find its way to the rivers.

Through the splendid work of the Engineer Corps of the United States army the first condition is being met by the

and, linked with this, the interchange of freight between railroads and river lines by means of through bills of lading.

Interesting as are the developments which have been made, and which must be made in future, to meet the first two requirements for river transportation, nevertheless it is the third requirement—the provision of suitable craft—which is of immediate interest to shipbuilders and transportation officials. It is this phase of the problem with which we propose to deal in this article.

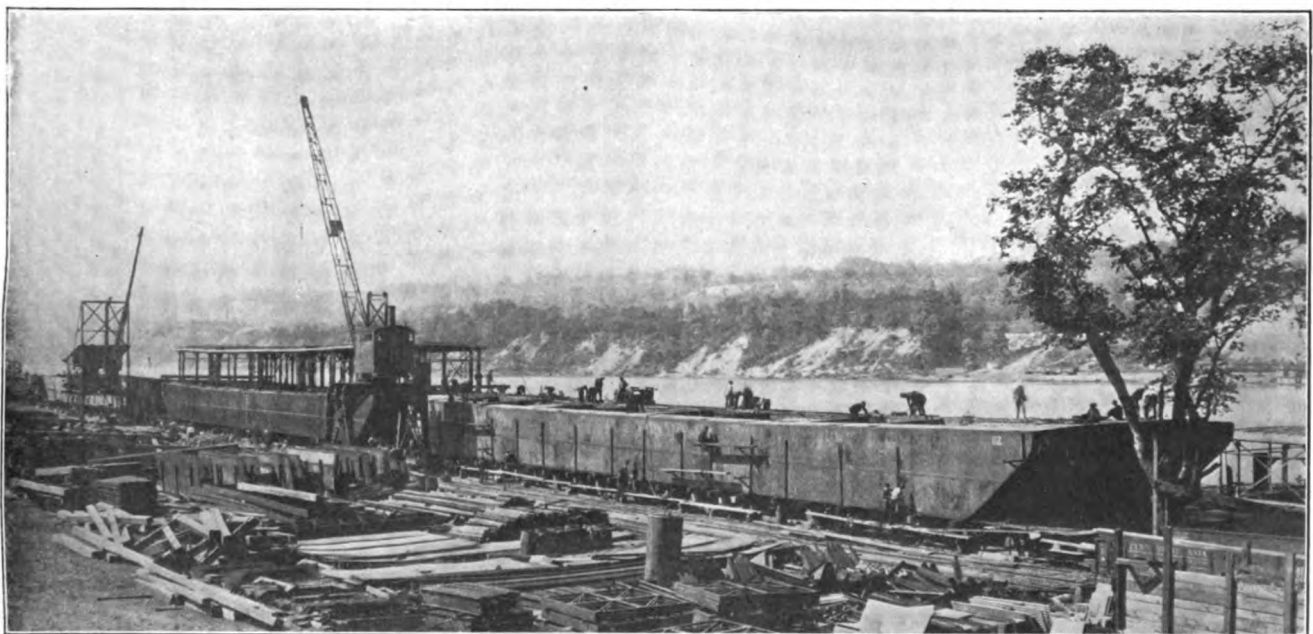


Fig. 1.—Shipyard at Ambridge, Pa., Showing Locomotive Gantry Cranes

United States Government as rapidly as the appropriations by Congress will permit. Especial attention has been given to the canalization of the Ohio, Missouri and Mississippi Rivers, while many other streams are being improved by dredging and by the protection of banks, etc. New canals are being built and others reconstructed, the largest single project of this character being the New York State Barge Canal. On the Southern rivers the development of water transportation has been greatly stimulated by the completion of the Panama Canal, and with this development there has naturally come an awakened interest in the construction of terminals for the rapid and economic handling of freight in bulk and package,

In the early days lumber was the material naturally used for the construction of river barges, both because it was usually available close at hand, and because its first cost was low with interest charges correspondingly so. The costs of maintenance and repairs, however, were high. Gradually good timber for barge construction became more and more scarce, until now it is almost unobtainable at satisfactory prices or of suitable quality. For this reason, and also due to the fact that improved river conditions and improved terminal facilities demand that barges and river craft shall be of more permanent and stable construction, the transition to steel has been naturally brought about.

Steel river barges were used successfully about ten years ago, although the first cost was then, as it is now, higher than that for wood construction, nevertheless the largely increased life of the craft, the reduced cost of maintenance, the lower operating expenses and the minimum cost for repairs, brought the ultimate cost of steel craft to a figure considerably lower than that for wood. Added to this is the very much decreased cost of insurance, not only for the craft but for the cargo, for insurance rates are now prohibitive on first-class freight in wooden barges, whereas they are extremely low for such freight when carried in well-constructed steel barges.



Fig. 2.—Steel Cargo Box Barge

To show the general features of design and construction which have been developed in recent years in the building of steel river barges for various purposes, we have secured some data from the American Bridge Company, Pittsburg, Pa., regarding the steel barges built recently at their barge and boat yard at Ambridge, Pa.

THE AMBRIDGE SHIPYARD

This yard, which is a department of one of the largest bridge and structural shops in the world, is located 16 miles west of Pittsburg, on the Ohio River. On account of its close proximity to the steel mills, and because most of the tonnage on the Mississippi River comes from this locality, the barge yard is ideally located for the manufacture of river craft. We use the term "manufacture" advisedly, as the material is punched, riveted and otherwise prepared by the best and most approved machinery before it is assembled.

By punching the plates with spacing punches, the rivet holes are perfectly alined, and the work of assembling, fitting and riveting is rapidly carried out. The material, which is fabricated in the large shops of the American Bridge Company, is delivered to the barge yard on railway cars, from which it is unloaded and stored and assembled by locomotive cranes of the gantry type, two of which are shown in the general view of the yard, Fig. 1. The barges are built on the steep bank of the river, and are launched according to Lake methods—side-wise from ways parallel to the river bank.

At this yard barges have been constructed for the United States Government for use in the improvement of the Ohio and Mississippi Rivers. Barges have also been built for transporting general merchandise or packet freight and for the transportation of such bulk cargoes as coal, sand, bulk oil, steel products and cotton. In addition, the output of the yard has included a number of car floats and the steel hulls for tow-boats, packets, dredges and pile drivers.

BOX BARGE FOR GENERAL CARGO

Fig. 3 shows the general construction and arrangement of a cargo box barge 200 feet long by 36 feet beam, with a depth of 10 feet 6 inches, eleven of which have been built in the last few years at this plant for the American Steel & Wire Company. The barges are used for carrying steel products of the subsidiary companies of the United States Steel Corporation

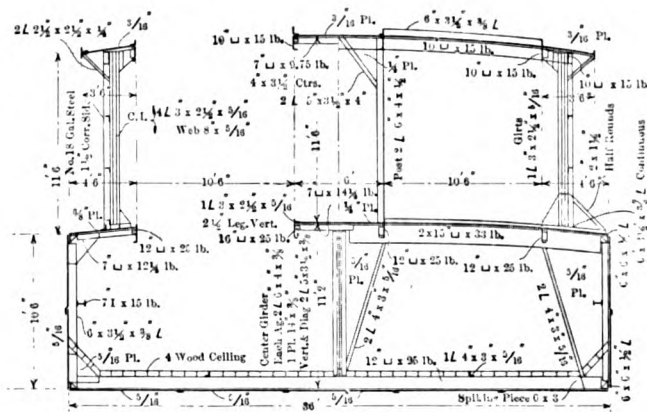


Fig. 4.—Midship Section of Cargo Box Barge

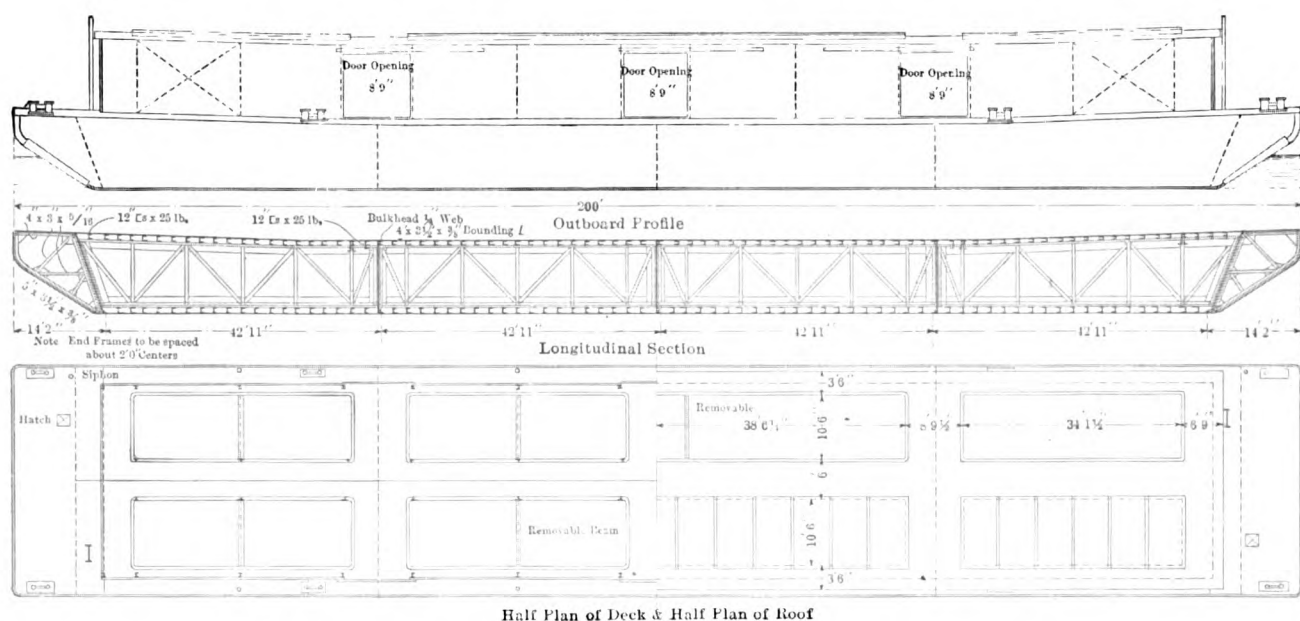


Fig. 3.—Profile, Longitudinal Section and Deck Plan of 200-Foot Cargo Box Barge

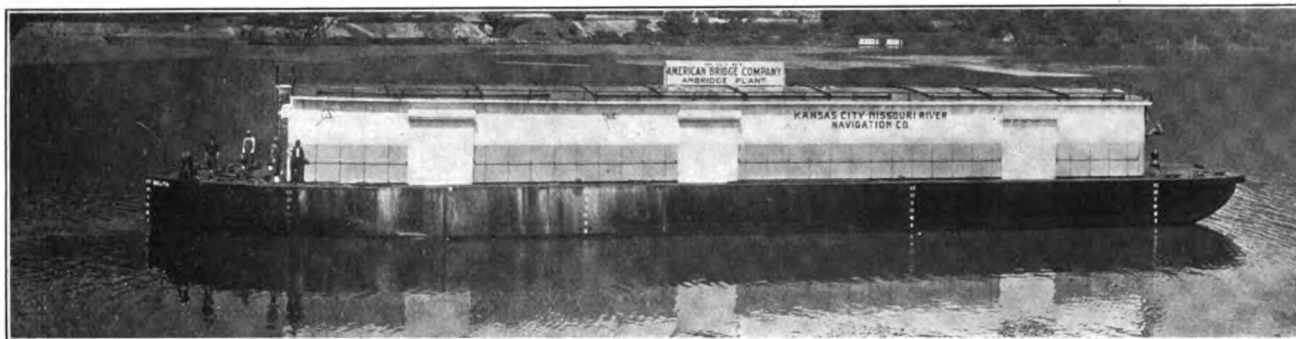


Fig. 5.—Barge Built for the Kansas City-Missouri River Navigation Company for the Transportation of Miscellaneous Freight

from the Pittsburg district to ports on the Ohio and Mississippi Rivers. The general construction of these barges is clearly shown by the photograph and drawings. The cargo box, which is 180 feet long, 31 feet wide and 11½ feet high, was necessary to give sufficient cubical capacity when carrying the lighter products, such as field fencing, barb wire, etc. Hatches were placed in the upper deck to permit the use of cranes in loading and unloading. These barges have a cargo-carrying capacity of 1,500 net tons on a draft of 9 feet.

Fig. 5 shows one of three barges which were completed last year for the Kansas City-Missouri River Navigation Company for the transportation of a general line of freight between St. Louis and Kansas City. By using this type of barge the insurance rates are sufficiently low so that first-class freight,

such as pianos, silk goods, etc., can be carried at minimum rates. The hull of this barge is 156 feet long, 30 feet beam and 8 feet deep. It has a rather full modeled bow and a type of stern known on the Western rivers as a "basket" stern. The deck is flat throughout without crown or sheer. Four transverse bulkheads divide the hull into five watertight compartments. Between bulkheads Nos. 1 and 4, and for a width of 12 feet each side of the centerline, the main deck is dropped 5 feet to form an inner bottom, the inner bottom and main deck being connected by inner side plates, thus giving the barge a double skin between bulkheads No. 1 and No. 4. The inner side plates and bulkheads Nos. 1 and 4 are carried up to a height of 10½ feet above the main deck to form the sides and ends of the cargo box. Bulkheads Nos.

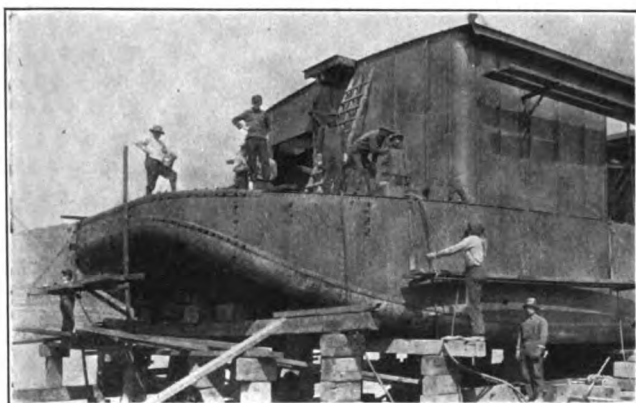


Fig. 6.—"Basket" Stern

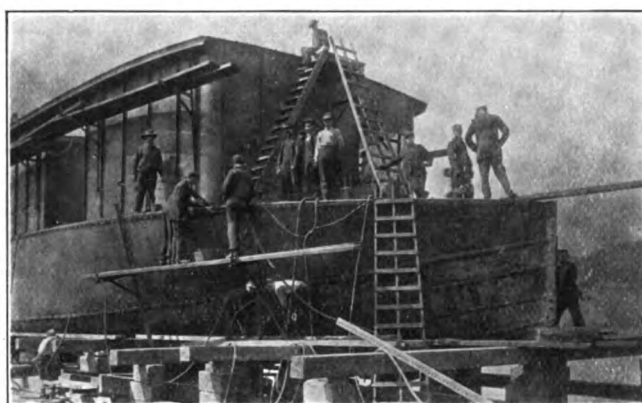


Fig. 7.—Form of Bow



Fig. 8.—Barge Loaded With General Cargo

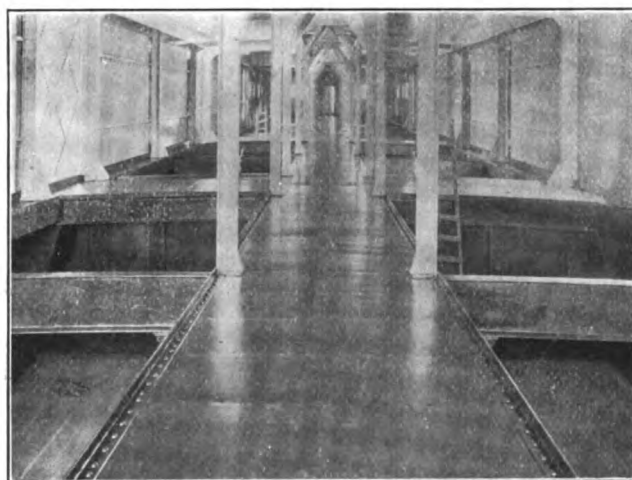


Fig. 9.—Interior of Steel Barge *Edgar Thomson*

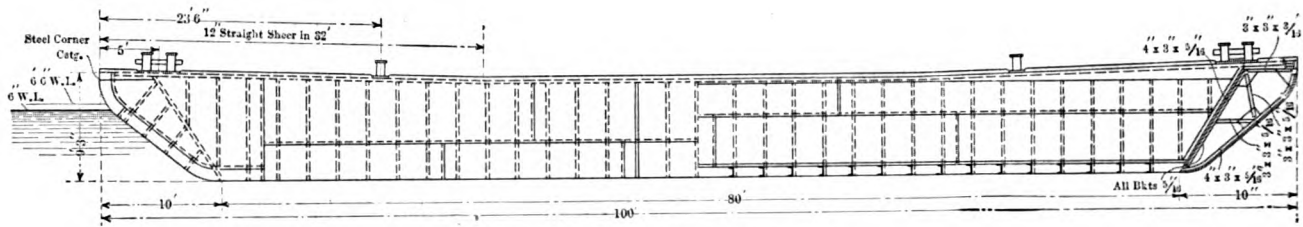


Fig. 10.—100-Foot Open Coal Barge

2 and 3 are also carried up to the upper deck to form fire partitions, being stiffened at the main deck by horizontal girders 3 feet wide.

A non-watertight keelson, 3 feet deep, runs between bulkheads Nos. 1 and 4. On top of this keelson, and spaced ap-

proximately 12 feet between centers, is a line of stanchions for the support of the upper deck. It will thus be seen that the cargo-carrying space in the barge is 24 feet wide, 15½ feet high and 120 feet long, divided into three compartments, the end compartments being 36 feet long and the center compart-

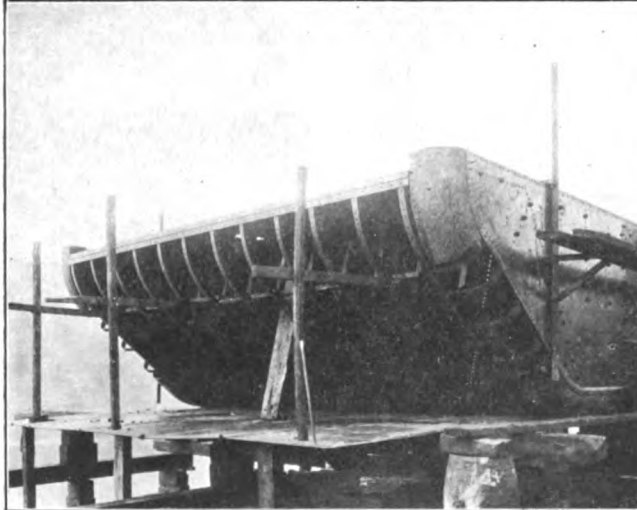
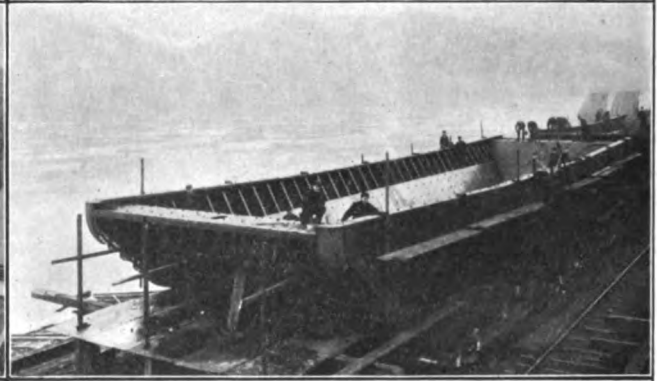
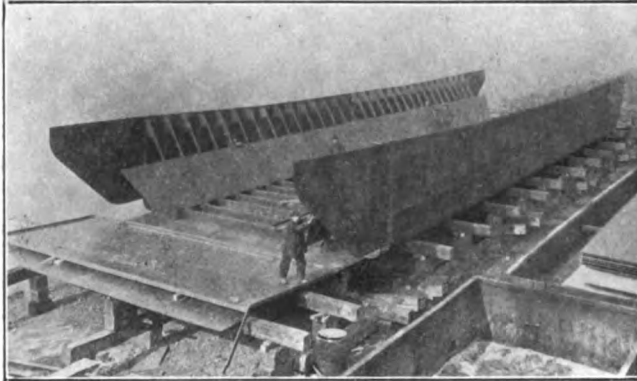
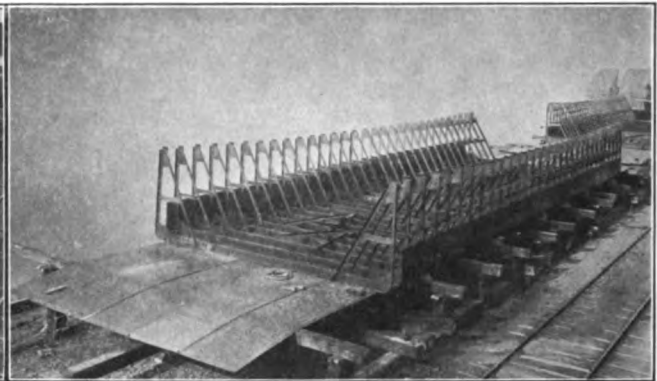
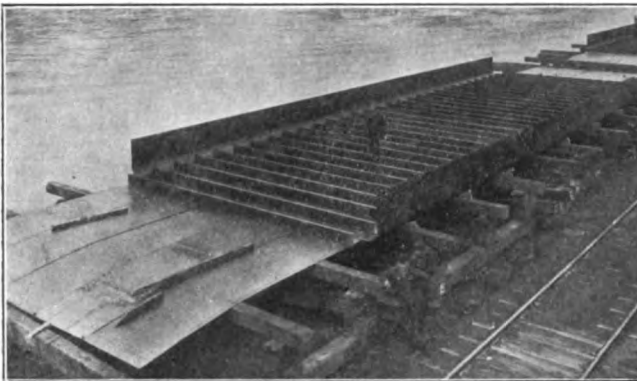


Fig. 11.—Floors and Bottom Plating

Fig. 13.—Side Plating

Fig. 15.—End Connections

Fig. 12.—A-Shaped Framing

Fig. 14.—Inside Plating

Fig. 16.—Interior of Finished Barge

By referring to the half-inboard profile, Fig. 10, it will be noticed that a very strong head log is provided. This is an essential feature in barges for river use, due to the fact that all barges are pushed ahead of the towboat and not pulled. A very heavy steel casting is placed at each corner of the barge and securely fastened to the side, knuckle and rake plates, because when entering the locks the tow is seldom parallel to the lock walls, and the corner of the foremost barge is subjected to severe abrasion.

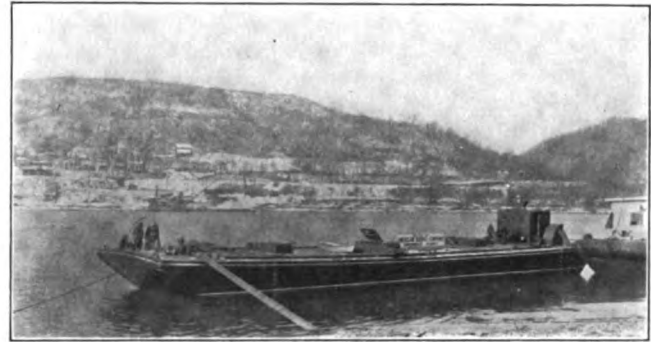


Fig. 19.—Oil River Barge

A decked sand barge, for use in connection with a suction dredge outfit, is shown in Fig. 18. Perhaps the most notable feature of this barge is the heavy side and deck construction. The sides of the barge are subjected to severe punishment on account of the method of handling. For instance, an empty barge is placed alongside the dredge, the lower or down-stream end being opposite the discharge pipe, and the barge is dropped down stream a little at a time as the cargo box is filled. The next barge to be filled is fastened at its forward end to the dredge, which lies up-stream from the barge being filled. The lower end of the second barge is held away from the dredge by the corner of the first barge. When the first barge is filled it is taken away, and the current swings the empty barge alongside the dredge in the correct position for loading. The heavy clam-shell buckets used in the unloading operation makes a very strong deck a necessity. At each loading the cargo box is filled completely, this quantity of sand weighing about 650 net tons, bringing the barge down to a draft of 7 feet.

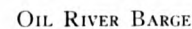


Fig. 19 shows one of three bulk oil river barges which were built recently from plans by and under the direction of Mr. George B. Drake, naval architect, New York City, for the Texas Company. In this barge there are three pairs of oil tanks having a combined capacity of 220,000 gallons. A complete system of in-take and discharge pipes is provided, the pump room being located in the after peak, together with all the valves. The steel deckhouse on this barge is equipped for the accommodation of a crew of four men.

This type of barge has a cargo-carrying capacity of 600 net tons on a draft of 6 feet. Fig. 8 shows the character of cargo ordinarily handled and the methods of stowing it in the barges.

Another type of barge built by the American Bridge Company, which is used to a great extent on the Monongahela River and in the harbor at Pittsburg for the transportation of coal and steel billets, is shown in Fig. 10, while Figs. 11, 12, 13, 14, 15 and 16 show clearly the successive steps in assembling one of these barges at the barge yard.

SHIPBUILDING RETURNS.—The Bureau of Navigation reports 127 sailing, steam and unrigged vessels of 20,052 gross tons built in the United States during the month of May.

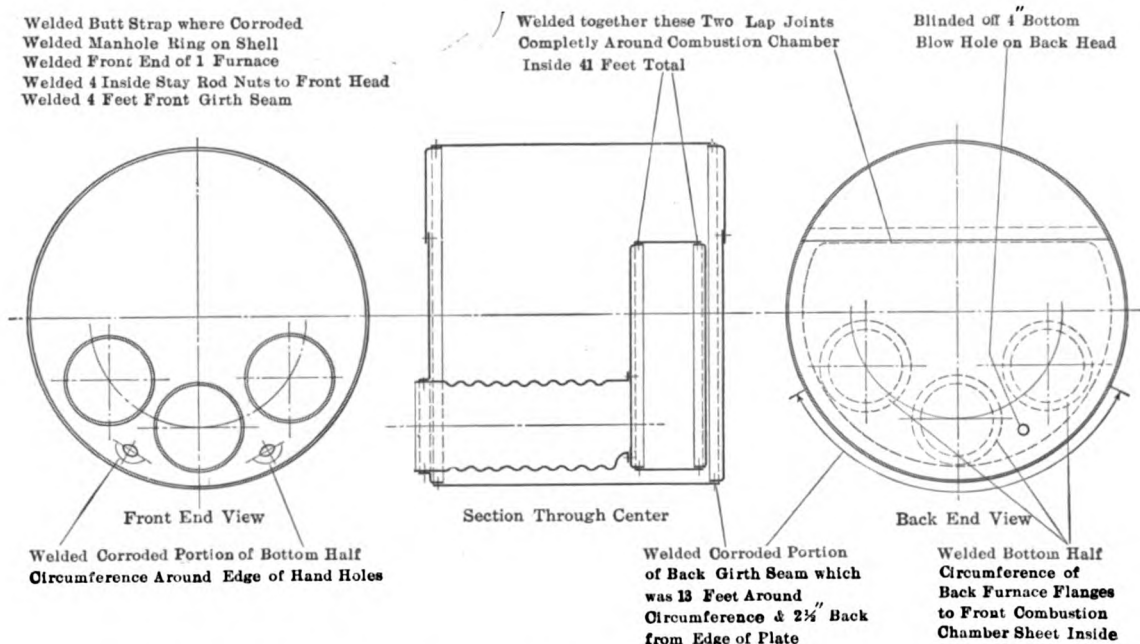
Repairs to Scotch Marine Boilers with Oxy-Acetylene Apparatus

Two boilers on the steamer *Meteor*, owned by the Pacific Coast Steamship Company, recently underwent repairs in Seattle, Wash. The work was done by the Olson-Klopf Welding & Cutting Company, Inc., with the Henderson-Willis oxy-acetylene process. On the port boiler 13 feet of calking edge was welded back on the head. One butt-strap

to be one of the largest repairs ever undertaken on boilers by the oxy-acetylene process.

Steamer Alvarado

The Craig Shipbuilding Company, Long Beach, Cal., has recently completed a lumber steamer called the *Alvarado* for the Long Beach Steamship Company. This vessel was built to carry lumber both in the hold and on deck, and certain



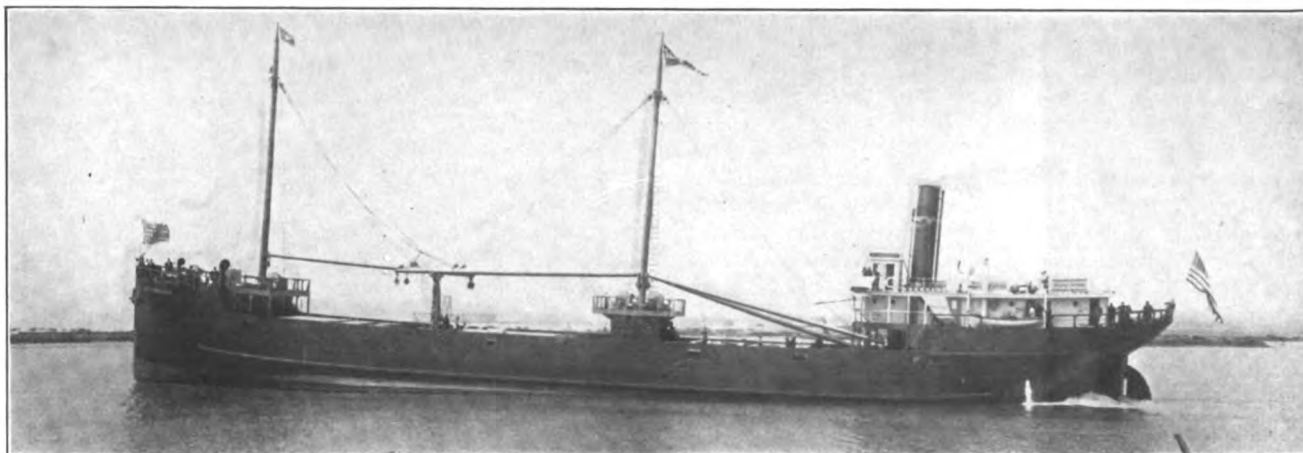
Welding Done with Oxy-Acetylene Apparatus on Two Scotch Boilers of the Steamer *Meteor*

was reinforced where corroded, one handhole and blow-off on the back head were reinforced, while in the combustion chamber 41 feet of calking edge, a patch, flame sheet, wrapper sheet and furnace end were welded. One manhole ring was welded and the front end of a furnace reinforced where wasted away. Three feet of girth seam were also welded. On the starboard boiler 12 feet of calking edge on the back head was welded, one butt-strap was welded and reinforced where corroded, one handhole and one blow-off in the back head were reinforced, 41 feet of calking edge was welded in the combustion chamber, and one manhole and 1 foot of shell on the front end of the boiler were welded. This is reported

of the holds are arranged to carry bulk oil. The following are the principal dimensions of the vessel:

Length overall	255 feet
Beam	42 feet
Depth, molded	16 feet
Gross tonnage	1,319
Net tonnage	940

Built to the Bureau Veritas rules for a single deck vessel, the *Alvarado* is also equipped to pass United States Steamboat inspection. She has a total lumber capacity of about 1,500,000 feet. Two steel masts, each fitted with cargo booms,



Twin-Screw, Single-Deck Steamer *Alvarado*, Designed to Carry 1,500,000 Feet of Lumber and a Cargo of Oil

the foremast with two booms and the mainmast with four booms, furnish ample means for handling the cargo. The cargo winches and windless are of the Craig Company's manufacture, while the capstan and steering gear are supplied by the Hyde Windless Company, Bath, Maine.

Propulsion is by twin screws driven by quadruple expansion engines with cylinders 11, 16, 23 and 34 inches diameter by 24 inches stroke. Steam is supplied by two Parker watertube boilers, in addition to which a donkey boiler is installed. Oil fuel is used under the Dahl system. The auxiliaries include an efficient electric lighting and oil pumping system.

The Loading and Steering of Lake Freighters

At the tenth annual meeting of the Pittsburgh Steamship Company officials, masters and engineers, held in Cleveland, Ohio, March 23 to 25, Mr. F. B. Smith, chief engineer of the line, read the following interesting report on the loading and speed of ships:

During the past season a number of questions have come up relative to loading of ships, speed of ships under different conditions, and in connection with the speed the question of steering has been brought out quite prominently.

Taking up first the question of loading the ships, and the hogging in some cases and in other cases sagging of hulls, we have gone into this matter quite thoroughly. In observing the loading of several of the large steamers, we found that beginning at the after end of the cargo hold, putting a pocket in every other hatch, and we went forward to within about the sixth hatch, and then going back to about amidships and coming forward with another run in order to get the ship's bow down, this plan of loading had no bad effect whatever on one of our modern ships, and after this run had been continued through, on taking the draft forward, aft and amidships, the boat was practically on a straight line; that is, she drew as much water at both ends as she did amidships and showed no signs of material sag in the center. It has been proved in the loading that if the boat is sagged in the middle to any material extent while loading, it does not improve matters any to put extra heavy loads in both ends; that is, the sag in amidships still remains until the boat is unloaded.

But in a paper read by Naval Constructor Stuart Farrar Smith, at a meeting of the Society of Naval Architects and Marine Engineers, in New York, December 11 and 12, 1913, Mr. Smith stated: That in practical experience the middle of the length of a long vessel may move up or down as much as 6 inches with reference to the ends, depending on the conditions of loading, without any material injury to the ship, and that a temperature rise of 1 degree Fahrenheit may cause a rise in the middle of the ship of as much as $\frac{1}{8}$ inch. Another paper was read by James E. Howard, engineer-physicist, covering a number of experiments of strains on ships' hulls, both at sea and while receiving cargo. In these experiments Mr. Howard used an instrument that would measure the stretching of both the deck and the side plates down to a point of 0.0001 inch, and by these tests he was enabled to establish accurately the vibrations caused by the revolutions of the engines separately from the vibrations caused by the weakness of the hull or by the sea.

It is a well-established fact that when the vibrations of the engine are synchronous with the vibrations of the hull, caused by the sea, the vibrations are greatly exaggerated and conditions may be materially improved by either changing the speed of the revolutions of the engine or by changing the propeller wheel. Also in Mr. Howard's experiments, he kept record of the changes in the form of the hull at different

times in the day, according to the temperature, and found that the apparent draft of the ship, taking the marks forward and aft, was changed as much as 2 inches, and in the discussion on these papers, it was brought out that the builders or contractors of government ships frequently find in certain classes of vessels that they can increase the apparent deadweight of the ship by even blowing off one of the boilers, and this is done sometimes in order to increase the bonus on their contracts.

We have had the question up before now of boats being loaded to certain marks at the head of Lake Superior, and when they arrived at the Soo they were drawing more water forward than they were when they left their loading port without having come up a corresponding amount aft. This has generally occurred when they arrived at the Soo during very warm weather, and in talking this matter over with a number of men who have made practical experiments in such matters, they expressed themselves very freely to the effect that this difference was caused by the sun heating the decks and the upper part of the hull of the ship, and that a partial remedy for this would be to turn the hose on the decks and cool them down. It was stated that if we would take a transit and make observations along the sides of the hull of one of our long steamers, we would find the change in the shape of the ship would be very considerable in the effect of different ways of loading and in the different temperatures in which these observations were taken. I have in mind at the present time a case which happened in Cleveland a number of years ago when the tall electric lighting masts were being built. The foreman in charge of the erection of these masts was somewhat startled on a hot morning when he noticed that the mast was quite crooked, but as the day went by he noticed the shape was changing, and in the afternoon the bend was on the opposite side. I mention this to show how much one of our ships would be affected by the heat of the sun on the decks while the bottom of the hull is submerged in cold water.

In the tests we made on the last three boats, which were put in commission during the season of 1913, we noticed there was a very marked difference in the speed of the ships when good wheelmen were at the wheel and when poor steering was being done. In one particular case, when Captain Watt was watching the steering in the pilot house and I was watching the wake aft and keeping track of the revolutions of the engine, steam pressure, etc., I noticed in one hour that the speed of the boat had dropped down nearly one-half mile, and also that the revolutions of the engine had been slowed up, but the steam pressure had been exactly the same; in fact, it had not varied as much as one pound during this period. I went forward to inquire how things were going there, and Captain Watt informed me that the wheelman had lost control of her and did some very poor steering for a period of 35 minutes out of the hour. Another man was then put at the wheel and he did very good steering, so that the bad steering, lasting only 35 minutes, had slowed the boat's speed down nearly one-half mile during that hour.

In connection with this, I wish to mention the statement made by one of the naval officers at the meeting of the Society of Naval Architects and Marine Engineers, in which he spoke of a government ship with bilge keels for the purpose of experimenting as to how much bilge keels retarded the speed of the ship. The bilge keels were removed after having made a trial run. Then another run was made over the same course with the same trim of vessel, and under almost exactly the same conditions, and it was found that it took more engine power to drive the ship at the same speed without the bilge keels than she had made with the bilge keels, but they discovered the cause, which was poor steering. The difference in the power was quite considerable, and the naval officer said it was a well-established fact that con-

tractors in running government ships on their trial trips were very particular to get a good helmsman, as the difference in speed attained in that way made quite a big difference with the amount which they received for their contract.

The mate on this steamer made it a practice of remarking in the presence of the wheelsman that nobody could steer that boat and keep her close to her course. I noticed, however, that before we got across the lake, and, in fact, for the last half of the distance across Lake Huron, the steering of the boat had been very much improved and also the speed of the boat had increased. We found later, however,

best real comparisons we have had, and demonstrated pretty thoroughly that where the boats are well steered, make good courses, and have the same kind of weather, there is practically no difference in their speed.

Going into the question of propeller wheels, I think we have demonstrated pretty thoroughly, not only on the tests made last summer but in a great many tests that we had made prior to that time, that in designing a propeller wheel the first thing to determine is the speed, or the number of revolutions per minute, that we wish the engine to make, and from this we must determine the pitch of the wheel that will give the best results for a ship of the type under consideration; then the area of the wheel must be decided upon and the wheel should be made as near as possible to a true screw, at the pitch that has been determined, and there is found to be a loss if the wheel is shifted from that pitch, either making it coarser or finer. If it is found that a mistake has been made in the dimensions of the wheel, it would be more economical to design a new wheel than to endeavor to correct the mistake by making a worse one in changing the pitch.

When the vessel is pitching in a seaway, throwing the propeller out of the water, there is quite a difference in the way different engineers handle their engines. Some of them watch

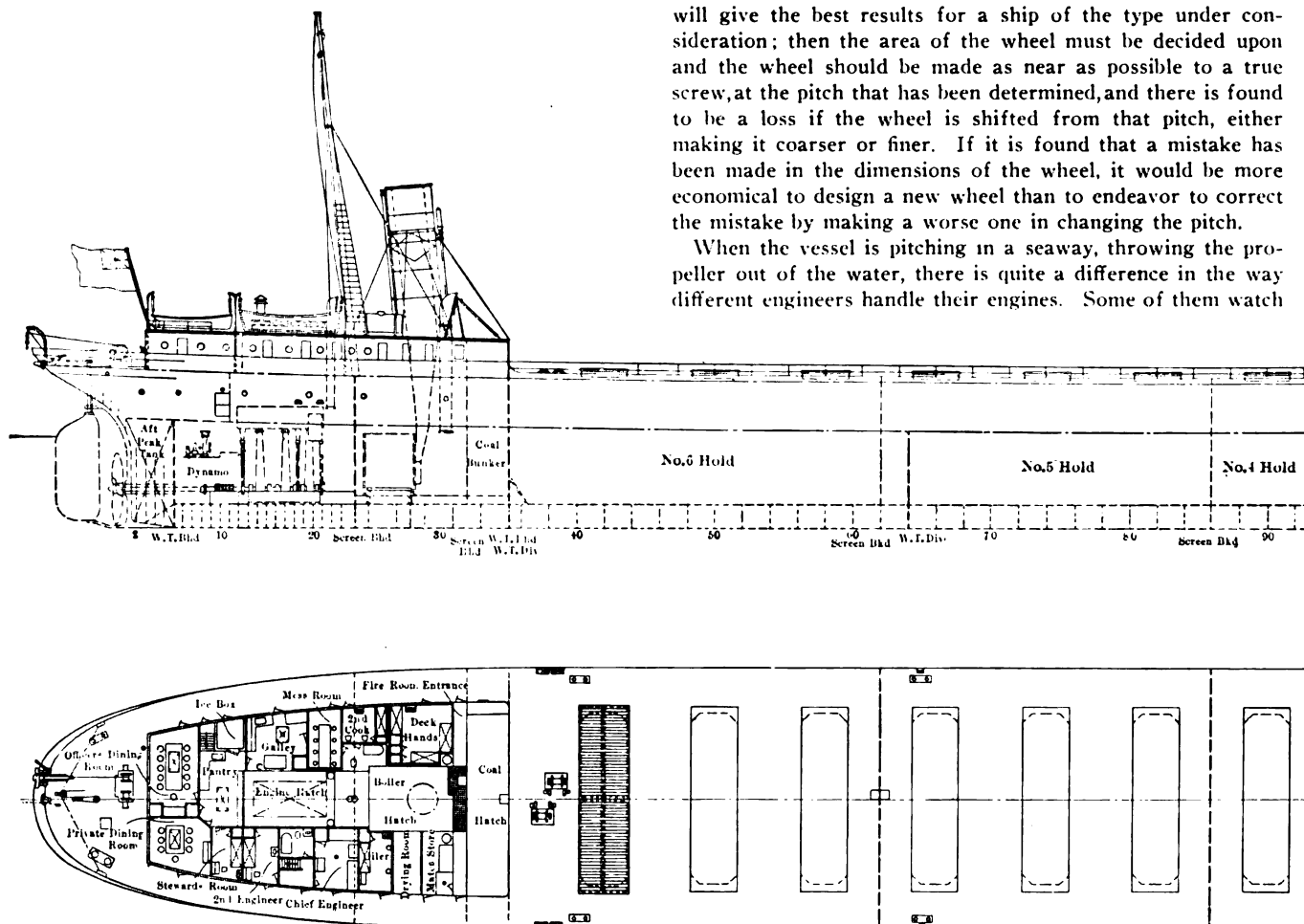


Fig. 1.—Profile and Deck Plan of 550-Foot Bulk Freight Steamer built

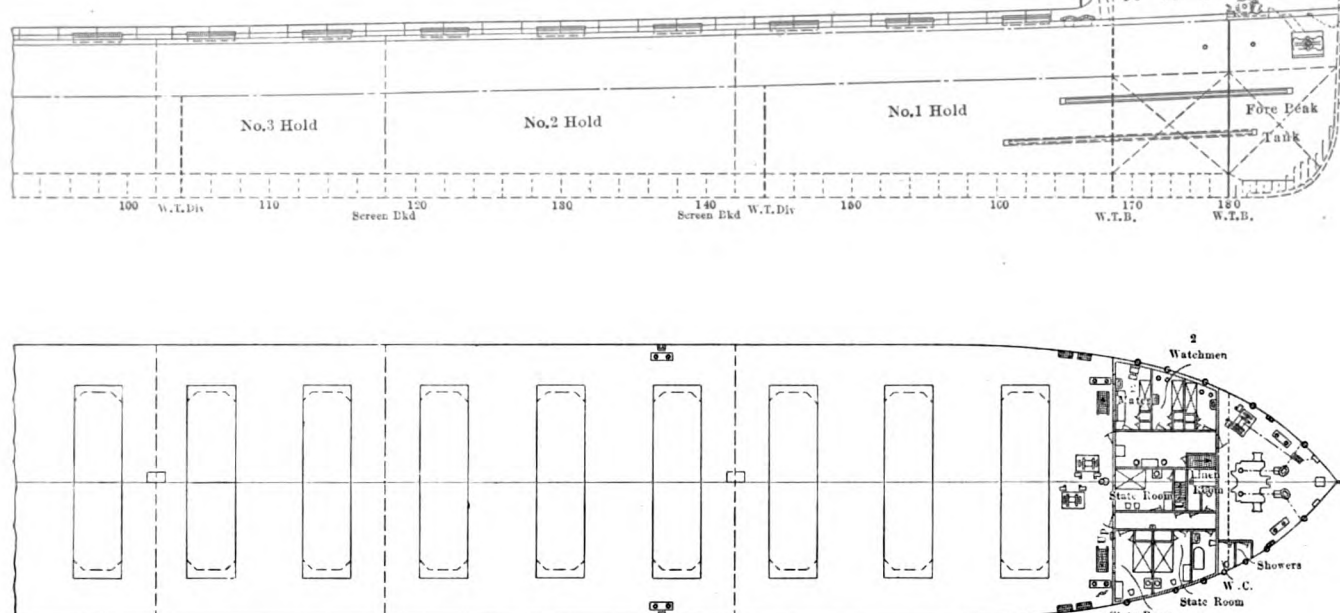
that there was too much lap on the controlling valve of the steering engine, and after that was corrected we were told that the boat steered better, and I saw no reason why she should not steer as well as either one of the new ships that came out last summer. Also, after those steamers were put in commission, we found there was a great difference in the speed of the three boats, but after making a run across Lake Huron on two of those ships, under almost exactly the same conditions of weather and trim, there was only a difference of three minutes in the time across Lake Huron, and only a difference of 135 revolutions. The weather was almost identically the same. There was only a difference of 0.4 of a revolution per minute in the two boats, and another very important point is that the boat that made the most revolutions in crossing the lake, and that had the highest revolutions per minute, was the one with the coarsest pitch of her wheel. Both of these boats on these runs were steered very fine, and by actual measurement on the chart made by Captain Watt there was only one-fourth mile difference in the distance they ran, so that this appears to be one of the

their throttling very closely, shutting off the throttle as the wheel comes out of the water, and the engine begins to speed up, and giving her full throttle again as soon as the wheel has become submerged, and in that way men who have given it special attention keep their engine running almost at a uniform speed; while with others I have noticed that they simply check their engine down in a seaway and do very little throttling, claiming that they make just about the same time. This has not been borne out in practice in any of the experiments that I have seen tried out. It has been my observation that those who watch the throttling very closely make better time in a head sea, and do it with considerably less strain on the engine as well as on the ship, and it occurs to me that this would be a very important matter for our older engineers to bring to the attention of the young men who are coming up in the business so fast; in fact, the handling of the engine is one of the things that I have noticed our young engineers are not as proficient in as the men who have been in the business for a long period of time.

Danger of Handling Heavy Weights in Cold Weather with Chain Slings

A pulley 30 feet in diameter with a 4-foot width of face had been cast in two halves. The joints had been planed, and the pulley was ready to be put in the pit lathe to be bored and machined on the face. There was very little head room to spare, and it was necessary in handling the pulley to use a chain sling around its face, as this gave a shorter bight than the heavy rope slings, and just allowed sufficient hoist to permit the traveling head of the crane to carry its weight out over the center of the face plate of the lathe.

Each half of the pulley had four arms, and it was to be chucked on the face plate of the lathe, which was about 12 feet in diameter, by suspending it on blocks placed under the arms and secured by bolts and straps to the face plate. One-half of the pulley had been placed in approximately the center of the face plate of the lathe, and the machinist was standing on the hub adjusting and securing the blocks, when the chain snapped; the pulley fell, and its overhanging rim striking the edge of the face plate, broke off big pieces of it and the



by the Collingwood Shipbuilding Company for Service on the Great Lakes

arms; the whole mass with the machinist was plunged down into the bottom of the pit a distance of 18 feet.

In falling the broken pieces of the rim struck large blocks that were close to the edge of the pit, and that had been used to chock the pulley when shifting the slings as it was being turned rim upwards, and these were carried down with the rest of the falling mass. The jar shook the building as though shaken by an earthquake, and all hands in the shop ran to the scene of the accident.

When the dust had cleared a little, a ladder was brought, and the machinist was taken from the pit, not crushed beyond recognition, as every one expected he would be, but with only a bad scalp wound, a bump on the forehead and a bad shaking up. The large blocks in falling in the pit had crossed in such a manner with the large, heavy pieces of the broken pulley as to keep them off the body of the machinist, a most miraculous escape.

The day was bitterly cold, and the doors of the shop being open while handling the pulley, the chain became crystallized and snapped under the strain.

J. E. C.

Great Lakes Bulk Freight Steamer

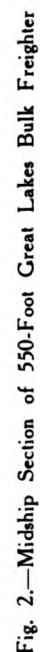
The Collingwood Shipbuilding Company, Limited, Collingwood, Ontario, have in hand at present a steel bulk freight steamer for the St. Lawrence and Chicago Steam Navigation Company, Limited, Toronto. The principal dimensions are: Length over all 550 feet, beam molded 58 feet, depth molded 31 feet.

The vessel is being built in excess of the highest class in the Great Lakes Register, and is of the single-deck, bulk-freight type, with side ballast tanks 5 feet wide to the height

of the main deck stringer. The construction is on the arch and web frame system, the arches and webs being spaced 12 feet apart, with channel frames between spaced 3 feet apart. A double bottom 5 feet deep is fitted for water ballast, extending all fore and aft, subdivided by a center keelson and transverse floors into nine watertight compartments.

The cargo space is subdivided into six separate holds by five transverse screen bulkheads. There are sixteen hatches spaced 24 feet centers, 38 feet by 10 feet, with steel coamings 15 inches deep and wood hatch covers 4 inches thick, supported on steel strongbacks, with double tarpaulins on top, all being rigidly fastened down by Mulholland's patent hatch fasteners spaced 3 feet 4 inches apart. In addition to this the hatch covers are also held down by strong angle iron hatch battens, securely connected by forged eye pads to the deck plating.

The vessel has a top gallant forecastle with crew's quarters and passenger accommodations. There is a steel deck-house on the forecastle deck, containing a stateroom, office and bathroom, for the captain, and a large observation room forward with access from captain's quarters, and a steel pilot



house on top. A steel deckhouse at the after end of the spar deck contains accommodations for the crew. Here also is located the galley, refrigerator and pantry and dining rooms.

A steam steering engine is placed on the main deck aft, geared to a semi-circular quadrant, with a spring buffer arrangement to take care of excessive strains on the rudder. Connection to the pilot house stands is by a rack and pinion transmission gear. An emergency steam steering gear is also fitted to the same quadrant.

The vessel is driven by a vertical triple-expansion marine engine, having cylinders 24 inches, 40 inches and 66 inches diameter by 42-inch stroke, capable of developing 2,400 indicated horsepower, supplied with steam by three Scotch boilers 13 feet diameter by 11 feet long, with a working pressure of 185 pounds.

The vessel is equipped with Howden's patent system of forced draft.

The Manchester, a New Harbor Express

The harbor express designed and recently placed in operation by the Lehigh Valley Railroad's floating equipment department has been built for the rapid handling of freight in New York harbor. It gives a maximum of service at a minimum of cost, and is proving particularly useful for the disposal of small consignments. It does, in a measure, the work an express wagon does on shore. It has been appropriately named the *Manchester*, as Manchester, N. Y., is the location of the Lehigh Valley Railroad's new freight classification station, which is a model of efficiency and economy.

The **Manchester** is 120 feet long over all; 32 feet breadth, molded; 16 feet depth of hold. Propulsion is by a 22-inch by 26-inch high-pressure engine, supplied with steam by a three-furnace Scotch boiler, 13 feet in diameter by 12 feet long. The freight house is amply lighted, an Engberg dynamo, 4½ kilowatts, supplying the power. A cluster of lights that can be carried from the boat into the car where freight is being transferred has also been provided.

There are a number of interesting details in the design and construction of the *Manchester*. There is no "drag" as in

at the after gangway practically as easy as at the forward gangway. In designing a boat with as much freeboard as the *Manchester* has it was necessary to get the weights as far down in the ship as possible. After the contract was placed it was decided to build a tank in the forward compartment in-

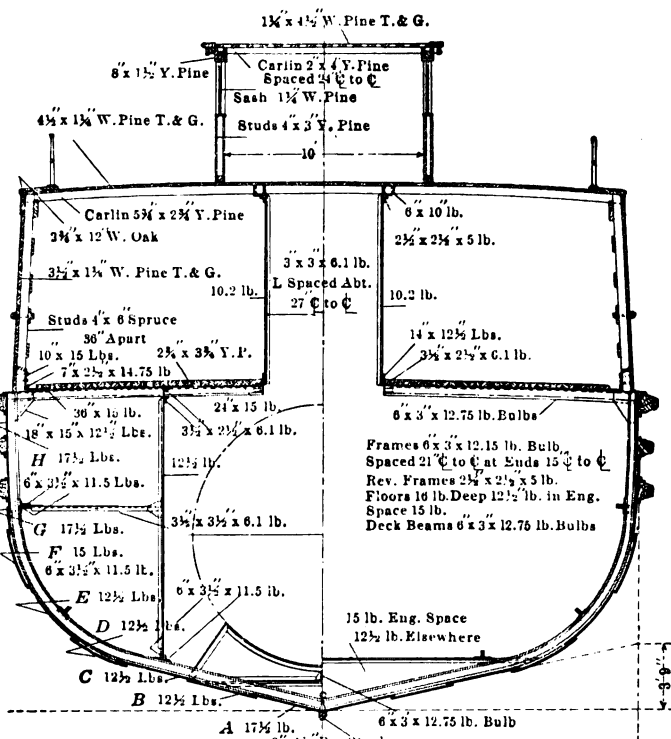


Fig. 2.—Midship Section

stead of building the round tanks as shown, or, in other words, to give the boat a double bottom. This was successfully done.

The location of the hoisting engine is on the upper deck, in order to give as much clear deck space as possible. The

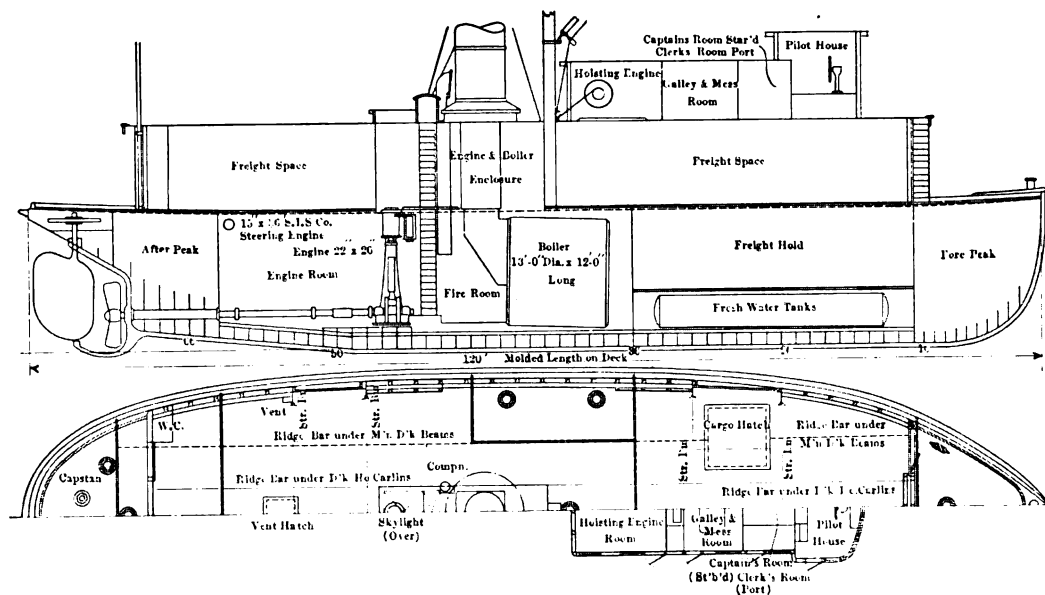


Fig. 1.—Longitudinal Section and Half Plan of the *Manchester*

the case of boats which trim by the stern, where the incline to the deck adds to the cost of handling freight. With 70 tons of coal and 40 tons of water the *Manchester* will have a free-board at the forward gangway of 7 feet 4 inches and at the after gangway of not less than 7 feet. This will make loading

amount of hoisting is also greatly reduced, as most of the freight is handled on trucks without the assistance of the hoisting. The height of the main deck is responsible for this saving, which will probably amount to about one-eighth of 1 percent a ton.



Fig. 1.—General View of the Bush Terminal, South Brooklyn, N. Y.

New Bush Terminal Pier

At the Bush terminals in South Brooklyn, N. Y., a new two-story pier, 1,400 feet long and 150 feet wide, has recently been completed and leased to the American-Hawaiian Steamship Company. The Bush Terminal comprises seven piers, of which the new double-deck pier is No. 6. These piers are

among the largest in the world, each being 1,400 feet long by 150 feet wide, the combined floor space of all the piers exceeding 2,000,000 square feet. The construction of the new pier was carried out under the direction of the Bush Terminal Company, and was completed in remarkably short time. The work began on July 23, 1913, and as the American-Hawaiian Steamship Company is to operate its fleet of twenty-eight

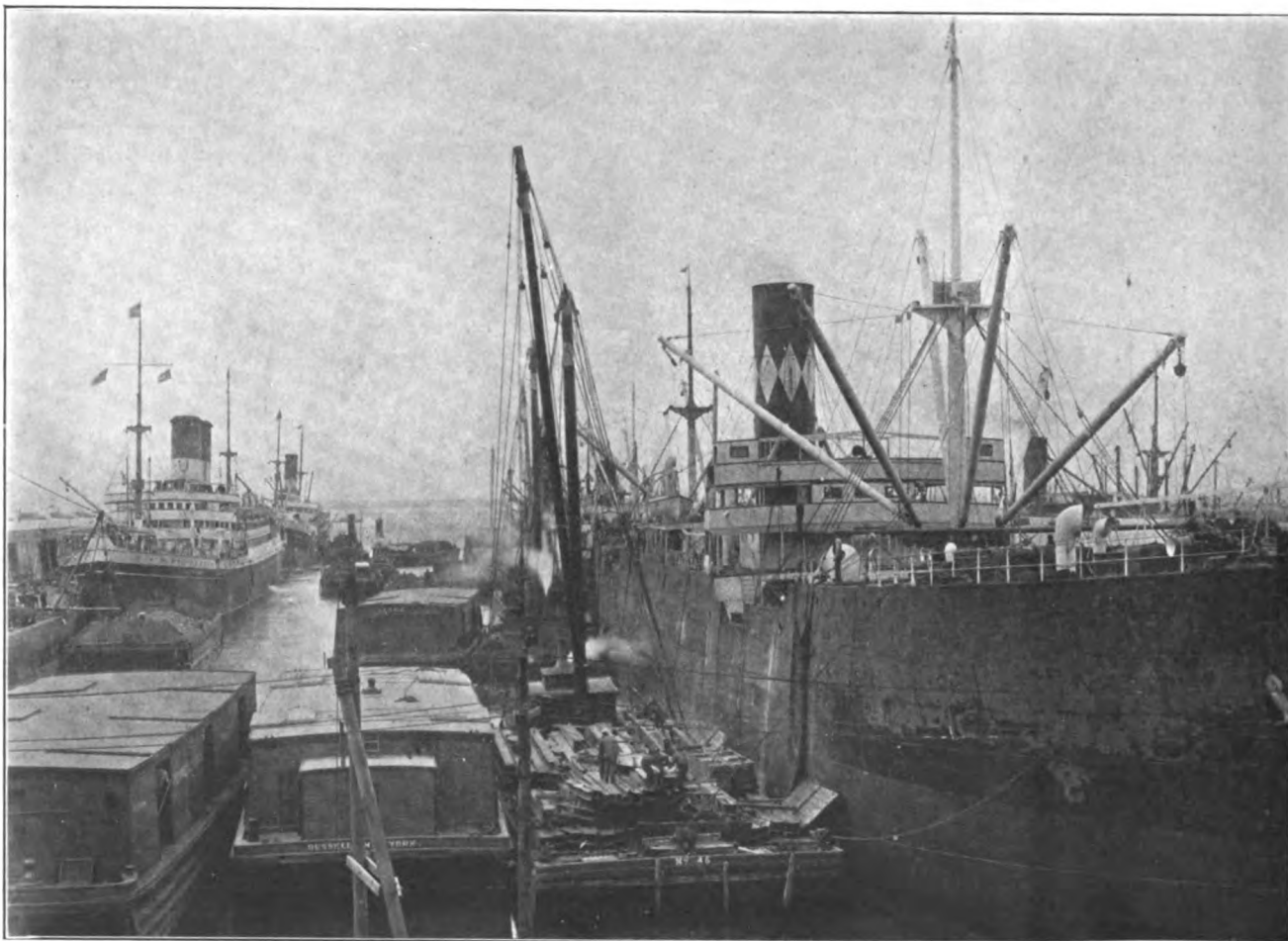


Fig. 2.—A Daily Waterfront Scene at the Bush Terminal

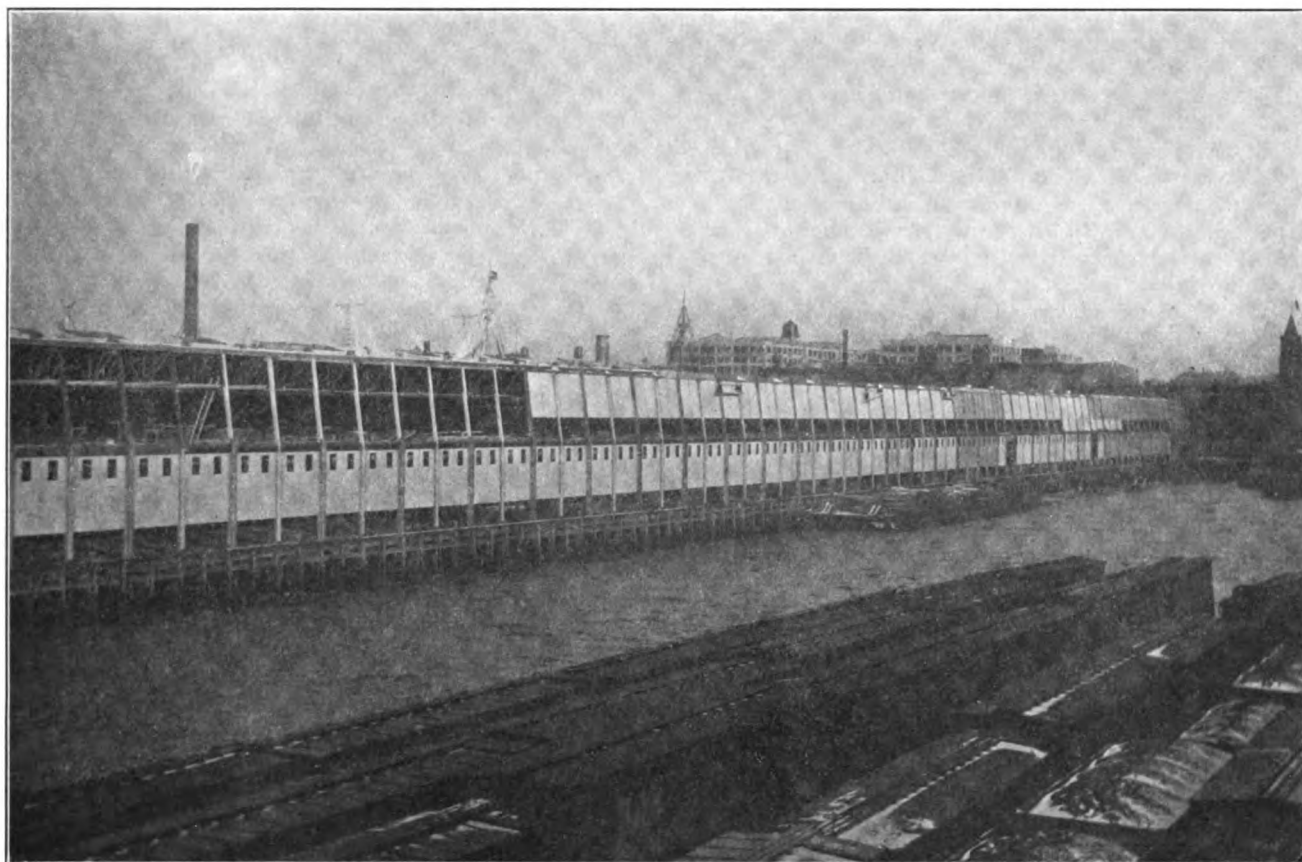


Fig. 3.—Side View of the New Double-Deck Bush Pier

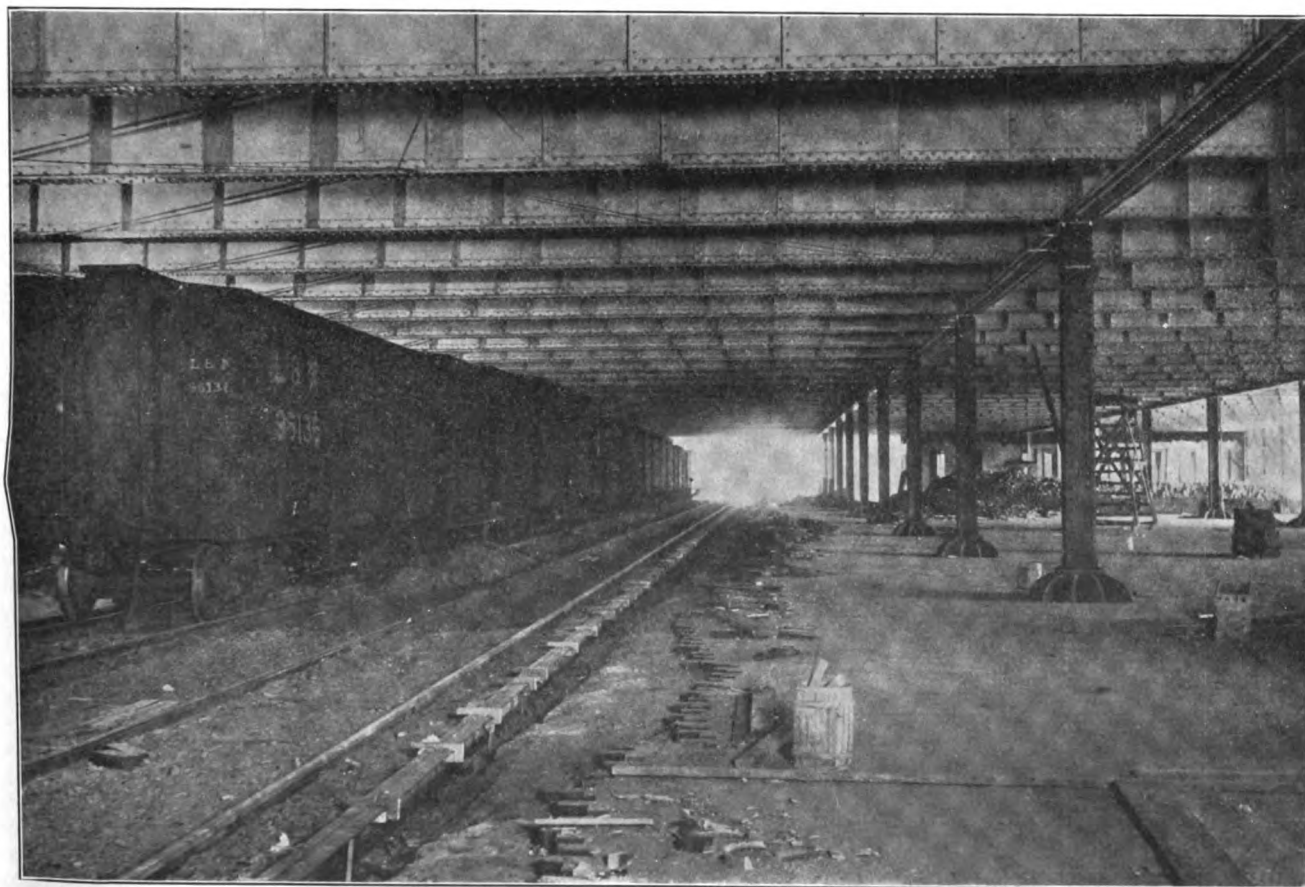


Fig. 4.—Lower Floor of the Pier, Showing Railroad Tracks in the Shed

vessels through the Panama Canal, every effort was made to complete the new terminal before the Panama Canal was opened for traffic. As a matter of fact, the pier was completed almost three months before the canal will be officially opened.

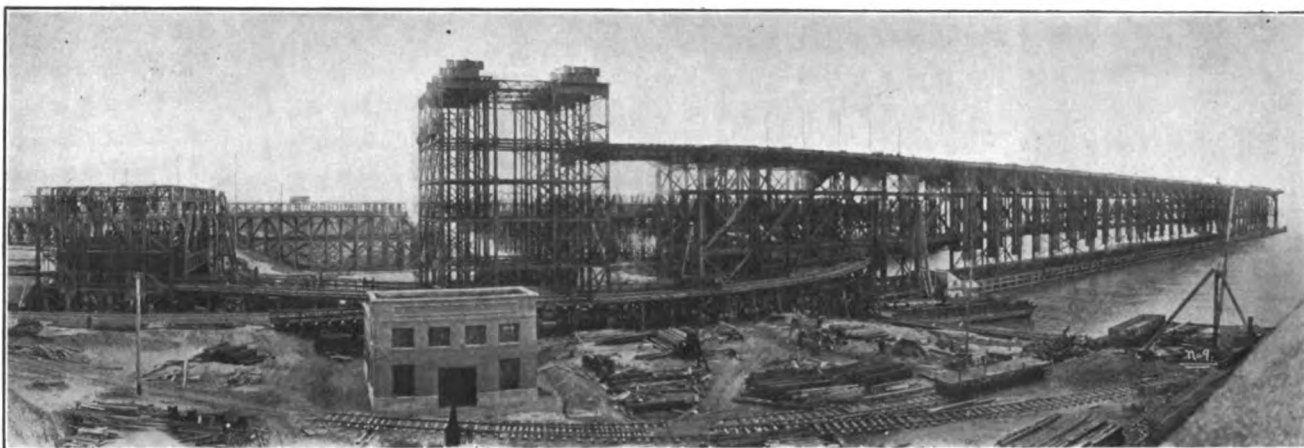
The handling of cargoes at the Bush piers is carried out along strictly modern lines. The width of the pier sheds relieves all congestion, and trains from all of the ten trunk lines terminating in New York City run alongside the piers, making possible the transfer of cargoes from the holds of the steamships directly into the cars and *vice versa* without the necessity of using lighters or other cartage. A space of 270 feet separates the piers from each other, and it is possible for ships to load and unload on either side of these slips, leaving sufficient leeway for the unobstructed passage of smaller craft used in handling the freight.

Pier 6 is modern in every aspect. The foundation piles were driven 30 feet below the low water mark, capped with

The Chesapeake and Ohio Railway Company's New Coaling Pier at Newport News

The new steel coal pier of the Chesapeake and Ohio Railway Company, recently completed at Newport News, Va., promises to exceed the world's present record for the rapid transfer of coal from the railway car to the holds of vessels. The plant has a maximum dumping capacity of about 5,000 tons per hour, which equals the capacity of any other coal pier on the Atlantic seaboard.

With one exception all other coal piers on the Atlantic seaboard are of the power incline type, in which road or transfer cars are pushed up an inclined plane to the top of the pier and dumped directly into bins at a maximum rate of forty cars per hour. The New Chesapeake and Ohio coal pier differs from this type of pier in having special transfer cars of 110 tons capacity, into which are dumped the roadway



New Chesapeake and Ohio Coaling Pier at Newport News, Va. Dumping Capacity over 5,000 Tons per Hour

concrete, floored with steel and cement, on top of which is a paving of chemically treated asphalt block. The supporting columns are of steel and the girders of the same material. The upper floor is of 4-inch lumber, covered with five layers of the heaviest tar paper, with an inch of yellow pine and another inch of maple on top. Melted tar is worked into the seams to make the floor absolutely air and watertight.

A double-track railroad runs along the entire length of the main deck. This is connected with the Bush Terminal Railroad, which in turn connects with all the big trunk lines. Electric and steam locomotives draw long lines of freight cars to and from the pier.

A 70-foot slip is cut in the head of the pier to permit lighters to berth without interfering with the big liners at the sides. The slips are dredged to a depth of 40 feet or more. Thus everything has been done to facilitate the movement of freight carried by water. Nearly 200 warehouses are backed up against the bulkheads, so that cargoes to be held in storage can be so disposed of without costly and damaging cartage, one of the biggest drawbacks to the use of many other pier locations in New York City and elsewhere.

OPENING OF THE CAPE COD CANAL.—The Cape Cod Canal, which extends from Buzzard's Bay to Cape Cod Bay, a distance of 8 miles, was formally opened July 29. The minimum depth of the canal is 25 feet and the width at the bottom 100 feet, while the approaches at each end of the canal, which have a depth of 30 feet, and which bring the total length of the new dredged waterway up to 13 miles, are from 250 to 300 feet wide at the bottom. The distance from New York to Boston via the Long Island Sound and the Cape Cod Canal is 260 miles, or about 70 miles less than the shortest route outside of Cape Cod.

cars; the transfer cars are then raised to the top deck of the pier by double elevators.

It was estimated that the dumper and elevators could be handled at the rate of thirty cars per hour for each dumper and elevator, but this speed has been considerably increased in practice, and the indications are that this rate can be increased to forty cars per hour for each dumper and elevator, thus materially increasing the capacity over original estimates. This combination of mechanical car dumper and gravity pier has been used for some time at the Virginian Railway dock at Sewall's Point, but the arrangement of elevators for raising the dock or transfer cars to the upper deck of the pier has never, with one exception, been used in this territory.

The Chesapeake and Ohio Railway Company handles all the coal delivered to vessels in the Newport News harbor. This coal is received from a large number of mines located on the company's lines in the New River and Kanawha coal fields of West Virginia and Kentucky, and is delivered to all classes of vessels, from the tug boat, requiring less than 50 tons, up to the largest government colliers of 10,000 tons or more tons cargo.

This coal is first classified in the main yards and then the cars are sent to the dumpers and elevators at the end of the pier, where the coal is dumped into electrically propelled dock or transfer cars, which move under their own power over a level track, passing from the dumper over the scales, to the elevators, which raise them to the upper deck of the pier. The cars then move again under their own power over a level grade on the top deck to the desired bin and after dumping their load they move to the outer end of the pier, passing through a switch or turnout onto a central track, and descend under air brakes and under control of an overhead trolley down a 64 percent incline to the level of the surface tracks,

passing through the dumpers, and then, by means of switches, they pass to a position under the dumper to receive a new load. A sufficient number of transfer cars are provided to provide for thirty operations of each dumper and elevator per hour.

The coal dumped into the pier pockets is held by gates until it is desired to chute it into boats anchored along either side of the pier. The top of the pier is 90 feet above high water, making it possible to handle coal at any height from 5 to 47 feet above high water. The slip on each side of the dock is 1,200 feet long and 225 feet wide, allowing four of the largest boats to be loaded at one time. The slips at the outer end of the pier have a natural depth of 50 feet of water and at the inshore end of the pier the berths are dredged for a depth of 34 feet of water, thereby insuring the easy handling of the largest vessels with their cargoes.

The Hulett dumper and elevator, built by Wellman-Seaver-Morgan Company, of Cleveland, is a double machine, each side having an estimated capacity of inverting thirty road cars per hour (this capacity, however, has been exceeded). The arrangement of the elevators, scales, dumpers and tracks being symmetrical on either side of the center line of the pier, gives a track on each side of the dumper on which the transfer car receives its load and a central return track for transfer cars returning from the top of the pier; properly located switches allow the empty transfer car to pass into the track alongside of either dumper for another load. The two elevators, having a lift of 75 feet (to the top deck of the pier, which is ninety feet above high water), are operated in steel towers about 105 feet high. The elevators are counter-weighted to equalize the pull on the machinery for the up and down movements of the platform and are electrically operated.

Electric power is used throughout in the operation of the pier. Each of the transfer cars has a load capacity of 220,000 pounds of coal. A sufficient number of transfer cars are supplied to provide for the operation of the dumpers and elevators to their full capacity.

The new pier has a larger number of spouts than any of the gravity or simple mechanical piers located on Hampton Roads. It is 22.3 feet higher than the Virginian pier and has as great a height as any pier on the Atlantic seaboard. The self-propelled cars are similar to those used on the Virginian docks, but of much greater capacity than any now in use, with one exception.

By the addition of this new mechanical gravity coal handling plant, the total rated capacity of the Chesapeake and Ohio terminal at Newport News is increased from 34,000 tons per day to 94,000 tons per day. The great advantage of this new pier to Newport News harbor will be the elimination of the loss of unnecessary time in loading vessels. While the rating of 100 tons per minute for this pier is unusually high, it has been demonstrated that the new pier is capable of developing this capacity.

Twin Screw Oil-Burning Tug Joseph Seep

The Greenport Basin & Construction Company, Greenport, N. Y., delivered to the Penn Mex Fuel Company, of Pittsburgh, Pa., on June 24, the twin-screw oil-burning tug *Joseph Seep* for service at Tuxpam, Mexico. The tug is 75 feet long overall, 18 feet 8 inches beam, with a depth of 7 feet, and a mean loaded draft of 4 feet 3 inches. The framing of the tug is of white oak, the planking of yellow pine and the decking of Oregon pine, while the hull is coppered for use in Mexican waters.

The contract for the construction of this tug stipulated that the vessel should be delivered within one hundred work-

ing days after the contract was signed. As a matter of fact, the tug was completed in eighty-seven working days, and will be delivered to the owners two weeks ahead of contract time.

Propulsion is by two 150-indicated horsepower inverted compound engines with cylinders 8 and 18 inches diameter by 12 inches stroke, supplied by the Portland Company, Portland, Me. The shafting is of steel, composition covered out-



Oil-Burning Tug Built in Record Time for Mexican Service

board, while the propellers are of composition. Steam is supplied by one Almy watertube boiler fitted to burn oil fuel on a combined Coen and National Transit Company system, with certain improvements as found desirable by the builders. Two cylindrical iron fuel oil tanks of 1,700 gallons capacity are installed alongside the boiler, while the water is carried in cylindrical iron tanks having a capacity of 1,300 gallons. The engines have an independent condenser, while the pumps are of the Blake and Kingsford Foundry & Machine Company makes. The system of communication between the pilot house and engine room was installed by Charles Cory & Son, and the electric light outfit was furnished by the Casey-Naser Electric Company, of New York. Thomas Drein & Son Company, Wilmington, Del., supplied the life-saving apparatus.

This tug was turned over to the owners complete ready for use and will proceed to her destination, Tuxpam, Mexico, by way of Galveston, in charge of Captain James H. Pierce, master, and Clarence J. Pierce, chief engineer. The speed of the tug on trial was 11.08 knots.

New Excursion Steamer Mandalay

An unusually interesting vessel has been placed in service this season on the Hudson River. She is known as the *Mandalay* and was designed by Messrs. J. W. Millard and Bro., naval architects, New York, for the Delaware and Hudson Steamship Company, New York. The hull of the boat is the hull of the former railway transfer steamer *Express*, the upper works of which were destroyed recently by fire. By slightly remodeling the form of the bow the hull has been rebuilt into a first class modern excursion steamer, typical of the luxurious river and harbor steamers which have been designed by Messrs. J. W. Millard & Bro.

The construction of the steamer was carried out by the Morse Dry Dock & Repair Company, Brooklyn, N. Y. In addition to the remodeling of the bow, the entire superstructure, including three joiner decks above the main deck, is entirely new construction. The main engines, condensers, thrust bearings, propellers, boilers, pumps, dynamos, steering engine and other auxiliaries were taken out of the hull of the *Express* and, with the exception of the main engines,

condensers, shafting, thrust bearings, donkey pumps and sea valves, a complete new outfit of machinery and boilers was installed.

The principal dimensions of the *Mandalay* are:

Length overall	283 feet
Length between perpendiculars.....	272 "
Breadth, molded	44 "
Breadth, over guards.....	61 "
Depth, molded	14½ "
Draft	9½ "
Gross tonnage	1,120
Net tonnage	407
Indicated horsepower	2,500
Speed, miles per hour.....	18
Number of passengers	3,000

and for supports to the windlass, capstan, chocks, cleats, etc. The whole main deck is laid with 3¼-inch Oregon pine. The engine and boiler rooms are encased by steel plating and a railing is fitted on the main deck around the cylinders of the main engines, which extend about 5½ feet above the deck.

Longitudinal stiffness and general rigidity have been obtained by steel bulwarks and houses on the main deck, together with steel stanchions, which have been fitted between the decks throughout. On the saloon deck is a large dancing floor and in way of this space a special trussing arrangement is provided in the form of longitudinal beams or trusses. This arrangement was adopted in order to eliminate so far as possible all obstructions to dancing. In this way only two stanchions are required on the dance floor proper.

The observation deck is in the form of a trunk deck with

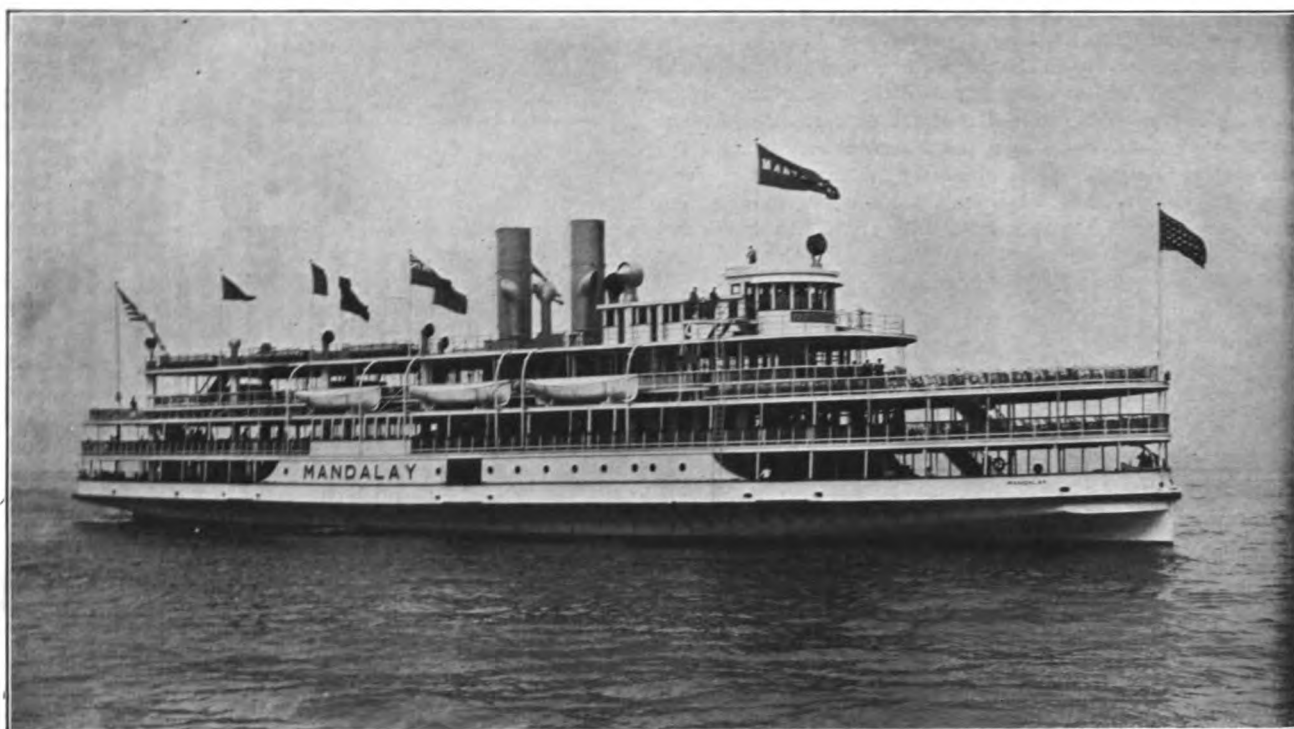


Fig. 1.—Twin-Screw, Open-Deck Excursion Steamer *Mandalay*. Capacity, 3,000 Passengers; Speed, 18 Miles per Hour

THE HULL

The form of the hull is very fine, conducive to good speed and yet the boat has proved remarkably stable on her trial and subsequent trips.

A bar keel, 8 inches by 2¼ inches, runs the full length of the hull. The frames are 4 inches by 3 inches by 7/16-inch angle bars, spaced 24 inches apart. Reverse frames, 3 inches by 3 inches by 5/16 inch, are fitted to each frame, while heavy web frames are spaced every 10 feet. Ordinary floors, 7/16-inch thick and 22 inches deep, are fitted in the bottom, while under the engines and boilers the thickness of the floors is increased. The side and bottom plating is 7/16-inch thick, while the sheer and garboard strakes are ½-inch thick.

Five watertight bulkheads divide the hold into six compartments. The bulkhead at the after end of the engine room has a watertight door for access to the propeller shafts. The main deck beams, which have a crown of 10 inches in 60 inches, are bulb angles 8 inches by 3½ inches by 7/16 inch connected to the frames with ordinary beam knees.

The hold forward of the boiler room is fitted with a tongued and grooved wood deck throughout the accommodations for the crew. Strips of plating are fitted to the main deck beams to take the coamings of the steel house

the sides stepped down for the storage of lifeboats, thus forming a separate boat deck and allowing an unobstructed view in all directions from the observation deck.

GENERAL ARRANGEMENT

Arrangements for the convenience of passengers and crew have been well taken care of. The galley and crew's quarters are on the lower deck, together with the officers' mess. These are very roomy and the galley is fully equipped with all necessary appliances.

An elaborate bar room and large toilet rooms for men, officers and crew are provided on the forward part of the main deck. The after part of this deck is taken up largely with the lunch room and its accessories. The usual dining room has been dispensed with and a buffet lunch counter installed.

Directly above this on the saloon deck is the dance floor, which is covered with ¾-inch linoleum and provided at the forward end with a slightly raised platform for the orchestra. Over the platform a sound well, or trunk, is provided so that the music can be heard by those on the deck above. Just forward of the dance floor is the hospital, retiring room and large toilet rooms for women.

The main saloon at the forward end of the saloon deck is artistically decorated in white with gilt trimmings. In this room is installed a very complete soda fountain, which bears the distinction of being the largest soda fountain ever installed on board ship. The floor of the main saloon is laid with "Dreadnought" tiling, making it not only substantial,

A very cosy and elaborately decorated observation room is provided forward on the observation deck. At the after end of this room a candy stand has been built in.

At the forward end of the hurricane deck is a large pilot house, containing all the necessary control appliances for steering and maneuvering the ship. A flying bridge extends

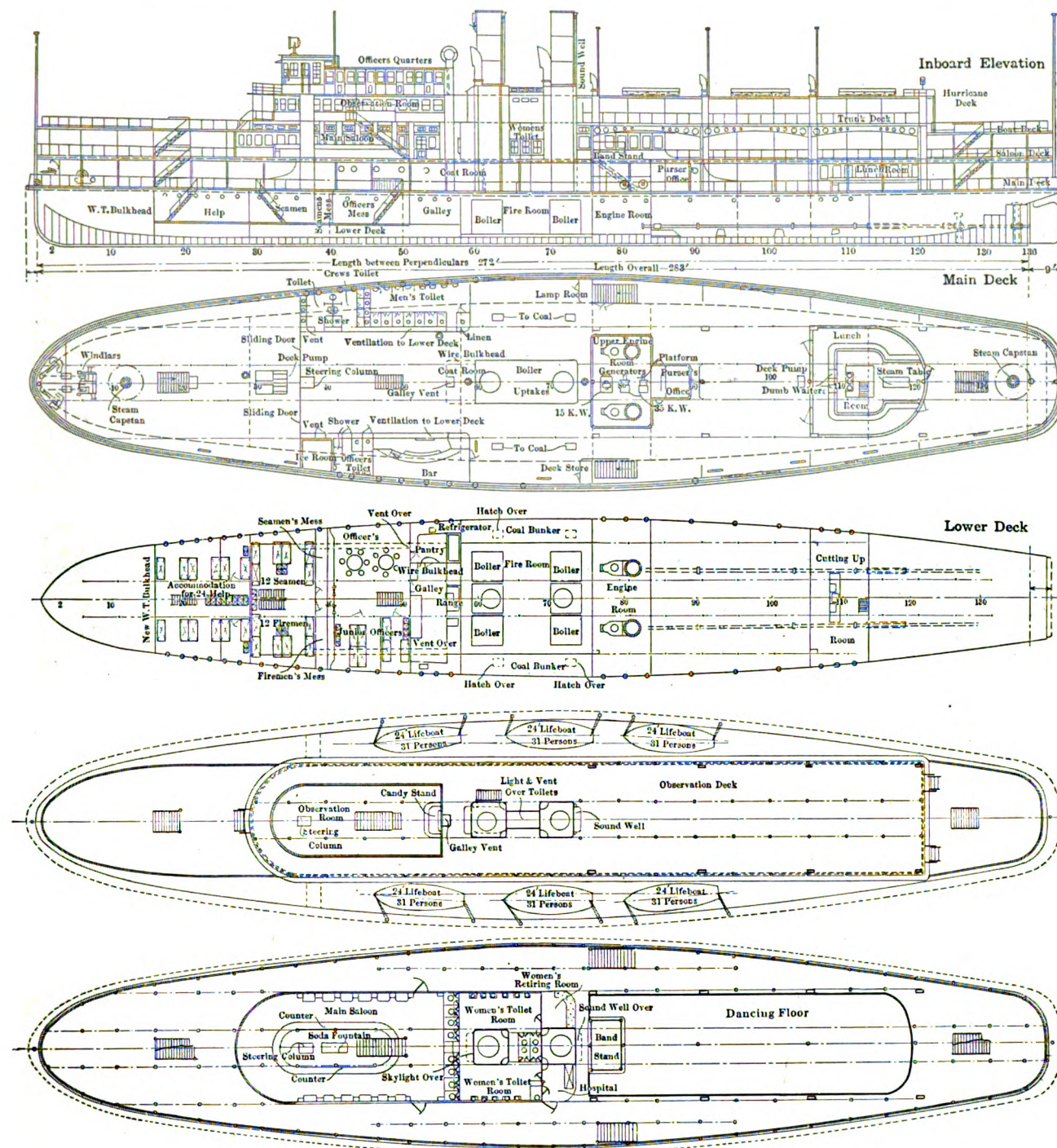


Fig. 2.—Inboard Profile and Deck Plans of the *Mandalay*

but attractive and easy to clean. Large, roomy seats are built in along the sides of the saloon.

As the *Mandalay* is designed for a large excursion boat, her decks are unusually spacious and, for the most part, are entirely open. In order, however, to cut off any strong wind, wind shields have been provided where necessary. These are in the form of large sliding doors of mahogany with plate glass panels.

athwartships on a level with the floor of the pilot house, while at the after end of the same deck are two smaller flying bridges, one on the port and the other on the starboard side, which enable the officers to maneuver the vessel with precision when docking.

Three searchlights are provided; one is a 30-inch projector on the pilot house, while the other two are 18-inch projectors located one on each of the after flying bridges. Aft of

the pilot house on the hurricane deck are accommodations for the captain, chief engineer and officers. These accommodations are remarkably complete in every detail.

"Nevasplit" panels have been used for deck paneling in the restaurants, saloon and hurricane decks, making the paneling not only decorative, but very strong and durable. A very good effect has been obtained by marking off the wainscoting in the main restaurant to represent large white tiles set with black cement, the wainscoting throughout and the flooring in all but the main saloon and observation room being of "Asbestolith," making them practically fireproof. The flooring in the main saloon and the observation room is done in "Dreadnought" tiling.

MACHINERY

The main engines, which were in the original hull, and which were built by the Harlan & Hollingsworth Corporation, Wilmington, Del., have been thoroughly overhauled under the direct supervision of Mr. E. P. Morse, Jr., at the Morse Dry Dock & Repair Company's plant. The engines are compound condensing with cylinders 26 and 48 inches in diameter and 36 inches stroke, developing altogether 2,500 indicated horsepower when running at 100 revolutions per minute. The condensers, which are of the surface type, with about 4,000 square feet of cooling surface, are built into the engine framing and act as columns supporting the cylinders.

Steam is supplied by six Almy watertube boilers arranged in two batteries of three boilers each, so that they can be

fired from the same platform, which is between them. The boilers are fed by two 9-inch by 6-inch by 10-inch Blake Admiralty feed pumps. The pumps are arranged so each battery may be fed independently or together. This same arrangement is also carried out in heating the feed water by the installation of a Reilly feed water heater with a set of twin coils. The coal bunkers are arranged amidships at each side of the boiler room and have a total capacity of 120 tons of coal, which is sufficient for about a week's supply.

AUXILIARIES

The auxiliary pumps for the fire, sanitary and drinking water systems were supplied by the International Supply Company. The pumps are arranged so that all can be connected to the fire system.

Electric current is supplied during the day by a 15-kilowatt generator directly connected to a vertical reciprocating engine, while at night two 35-kilowatt Curtis turbine generators are placed in service. The entire electric installation throughout the ship was furnished by the General Electric Company, Schenectady, N. Y. About 1,000 lights are installed, together with the three searchlights mentioned above.

The vessel is steered by a hand and steam steering gear of the Williamson type. There are also Williamson steam capstans and windlasses. The lifesaving appliances include six lifeboats and six liferafts, all of which can be handled easily by six sets of davits. The davits are of solid steel forgings made at the Morse Dry Dock & Repair Company's shops.

Care of the Electric Plant on Board Ship

The Principal Faults to be Found in Wiring—Points on Buying, Laying and Inspecting Cables, Wires, Etc.

BY SIDNEY F. WALKER

Cables and connecting wires of any electric light or power plant, or any other installation using electricity, form the weak portion of the plant. In the first place, the old caution should be given: do not cut the cost of the cables. Of course, you are to get the very best value for the money that is spent; but assuming you get good value, the money spent in the insulation and protection of cables, and of connecting wires generally, will repay itself over and over again. Put it in another way. Five percent or 10 percent saved in the cost of cables, providing that good value is obtained with the larger sums spent, may lead easily to the expenditure of ten or twenty times the amount saved, besides the great inconvenience involved.

Cables and connecting wires are subject to two principal faults. The smaller ones on board ship are subject to being parted, to the conductor being severed, and the supply of current to the apparatus to which the wire is leading being cut off. The larger cables, even very large cables, are occasionally subject to the same trouble; but cases of this kind would necessarily be rare.

Both large cables and connecting wires of all kinds, down to the very smallest flexible cord, are liable to short circuits, whole or partial, due to defective insulation. The insulation may become defective through the insulating material in the first instance being inferior, through the cables being carelessly laid, or through accidents, such as are common to every apparatus that is fixed on board ship. When even the strongest built ship is "working" in a heavy seaway every part of her is "working" more or less, is more or less in motion, and the results of the working and the motion may lead to the insulation of the cables or connecting wires being damaged.

In laying cables great care should be taken that where they have to make elbows to go round curves, say when going through the watertight subdivisions, provision is made for the expansions and contractions that go on with the changes in temperature, which seamen know only too well, and for the strains due to the changes of position when the ship is working heavily in a seaway. Provision should also be made to withstand the vibration to which the whole ship is subject when driven by reciprocating engines. Everyone on board becomes used to it; the old craft herself becomes used to it. Notwithstanding all that the vibration does its work, and the great danger is that one or the other of these causes, sometimes working separately, sometimes working together, may damage the insulation. The insulating covering of all cables and connecting wires is mechanically weak. All electrical insulating materials are mechanically weak. That is one of the great difficulties the electrical engineer has to contend with. He must provide insulation under all sorts of conditions where mechanical strength is also of the utmost importance, and he has the greatest difficulty in doing so.

There is the usual difference of opinion among electrical engineers as to whether armored cables should be employed on board ship or not. Armoring has the advantage that it will resist some of the strains that have been referred to if the armor has been properly put on. On the other hand, the armor sometimes passes on a squeeze. Modern cables are very much better constructed than those of twenty years ago; but there is still the possibility that in the process of laying on the armor, when the cable is passing through the armoring machine, a certain amount of squeeze may be given to the insulating envelope at a certain weak point. The

squeeze will not be found out until a long time afterwards; but it is like a man having a weak chest, or a weak organ of any kind, when the strains come on it finds out the weak spot.

A caution should be given here in buying cables if india rubber is used as the insulating envelope. The writer believes that it is not so common now to employ inferior African rubber as it was some ten or fifteen years ago. African rubber, as put upon the market to-day, is better than similar rubber that was on the market before the great development of rubber planting. As late as ten years ago African rubber was worth only one-fifth as much as Para, Brazilian rubber, and when made up into the insulating envelope of a cable there was no test that would discover the difference. The test came, however, not very long after. Where a cable made with a proper portion of Para rubber would last, say, ten years, those made with African rubber would often not last one year.

There is the usual controversy also as to whether rubber, bitumen or paper is best for the insulation of cables for use on board ship. At present the writer prefers rubber, good rubber, with 30 percent of Para; the insulating envelope being so made up by vulcanizing, and by properly preparing the different layers, that the final product is a continuous tube, closely encircling the conductor. And he likes the rubber-insulated conductor, after it has been protected with some form of jute wrappings, impregnated with waterproof material, drawn into a flexible lead tube of fair thickness. The lead tube, in his opinion, protects the insulation better than armor, and it is not so liable to pass on any squeeze that comes. It protects the insulating envelope by accepting the squeeze in its own body.

Bitumen has been largely used for board ship work; but it has the objection that in hot climates it naturally tends to soften, and it is liable to cracks with the working of the ship, with changes of temperature, etc.; while if cracks develop the salts from the sea will find their way through the crack to the conductor, and in the case of small conductors will part them.

The old rule that has been given for other apparatus may be repeated in connection with cables. Examine every inch of the cable and of all connecting wires as often as you possibly can. In particular, look out for signs of heat, and for that green powder, also for rust where armor is used. Where a lead covering to rubber cables or to paper-covered cables is employed look out for a white powder. Look out for cracks in lead tubes and in bitumen. The green powder is a salt of copper, an oxide or a chloride, and it denotes that the salts from the sea have got at the copper conductor. Rust, of course, shows that it has got to the iron. The white powder shows that the lead is being attacked.

One important caution may be given here. In the case of a lead-protected cable, particularly bitumen insulation, protected by lead; if there is a crack in the lead and a crack in the bitumen inside, as will probably follow, moisture impregnated with salts from the sea will almost surely enter, and there are then the conditions necessary for galvanic action. The copper and the lead and the moisture with the salts form a galvanic battery in themselves. In addition, the electrical pressure being always on the copper conductor there will be a leakage current passing out through the crack, through the moisture, tending to gradually eat away the copper where the conductor is positive, and to gradually eat away the lead where the conductor is negative. In the latter case further secondary action will probably be set up.

Where paper-insulated cables are employed it is of the very utmost importance to examine the lead coverings as frequently as possible. Any crack, or even a pin-hole in the lead, allows moisture from the atmosphere, which is always present between decks in every ship, to penetrate to the insulating en-

velope. What happens afterwards is usually as follows: The moisture creeps along inside the capillary space which exists between the paper and the lead tube, and it finds out some places in the paper where perhaps it is not fully impregnated with oil, and it proceeds to work its way through these. Gradually the insulation resistance of the cable is lessened and lessened at that point, and at some awkward, inconvenient time a spark passes across from the lead to the cable, or from the cable to the lead, and you have a short-circuit between the two.

As mentioned in a former article, also, flexible cords should be very frequently examined. They are the weakest spots of all, and particularly on board ship. Examine them for kinks; examine them for pieces of wire sticking out through the covering; examine them for the deposit of moisture and of the salts from the sea upon any part of them. The flexible cord is a very useful thing indeed. It enables lamps and other apparatus to be employed under conditions that would be impossible without them, but it is a source of great danger. In the first place, if it is of inferior quality when fixed, and in the second place if it is not properly looked after, if it is not frequently examined. The modern flexible cord is again far superior to that of twenty years ago, but it is still liable to the troubles mentioned. It should be remembered that a "short" in a flexible cord may lead very quickly to a fire. The cord itself may become red hot in a few seconds, and almost anything may happen after that.

Design of Self-Propelled Barge for the New York State Barge Canal

In response to the offer made in the June issue of this journal to publish designs or suggestions from our readers which will aid in establishing the best type of barge for service on the New York State Barge Canal, the following information has been submitted by John H. Bernhard, of New Orleans, formerly general manager of the Alabama & New Orleans Transportation Company, which is now operating a fleet of 1,000-ton coal barges for bringing coal from the Alabama coal fields to New Orleans. This project was conceived by Mr. Bernhard, and it was through his efforts mainly that the enterprise was carried out.

The route over which the barges of the Alabama & New Orleans Transportation Company are operated is from New Orleans up the Mississippi River, then through the Lake Borgne Canal to Mississippi Sound, thence through Mobile Bay to the Tombigbee River, up the Tombigbee River to its junction with the Warrior, and up the Warrior River to its junction with the Black Warrior, which is followed to Tuscaloosa, Ala., the terminus of the line. The entire distance is about 500 miles, and the river distance from Mobile to the head of navigation is about 444 miles.

This route has a striking similarity to the new inland waterway from New York to Buffalo, which is up the Hudson River and through the \$130,000,000 (£26,650,000) New York State Barge Canal, including Oneida and adjacent lakes. The Mississippi River might be compared with the Hudson River, while the Lake Borgne Canal, whose channel is 8 miles long, 80 feet wide and 8 feet deep, offers the same difficulties to navigation as the New York Barge Canal, with the added limitation that the channel of the Lake Borgne Canal is more crooked.

The Mississippi Sound from Lake Borgne to Mobile Bay might be compared with the lakes which the New York Barge Canal traverses. Then come the Mobile, Tombigbee, Warrior and Black Warrior Rivers, which have been made navigable by the Government after twenty years' development at an expenditure of \$8,000,000 (£1,640,000). In this route



Fig. 1.—Alabama Coal Barge Under Way in Mississippi Sound

there are seventeen locks, each 52 feet wide and 285 feet long, most of which have a rise of 10 feet, while one lock has a rise of 63 feet. This river is extremely crooked and difficult to navigate, yet the Alabama & New Orleans Transportation Company, which undertook to move coal from the head of navigation of the river to New Orleans, a distance of 515 miles, has built seven self-propelled barges with a capacity

of 1,000 tons each, which have been operated on this route very successfully.

Fig. 2 gives a good view of this river at Demopolis, Ala. In this picture can be seen one of the company's barges moored against the shore unloading salt, while in the background can be seen an old-fashioned stern-wheeler. Difficulties have been encountered in navigating this route with stern-wheel vessels



Fig. 2.—Barge Unloading at Demopolis, Ala.



Fig. 3.—1,000-Ton Barge Going Through Lock in Lake Borgne Canal

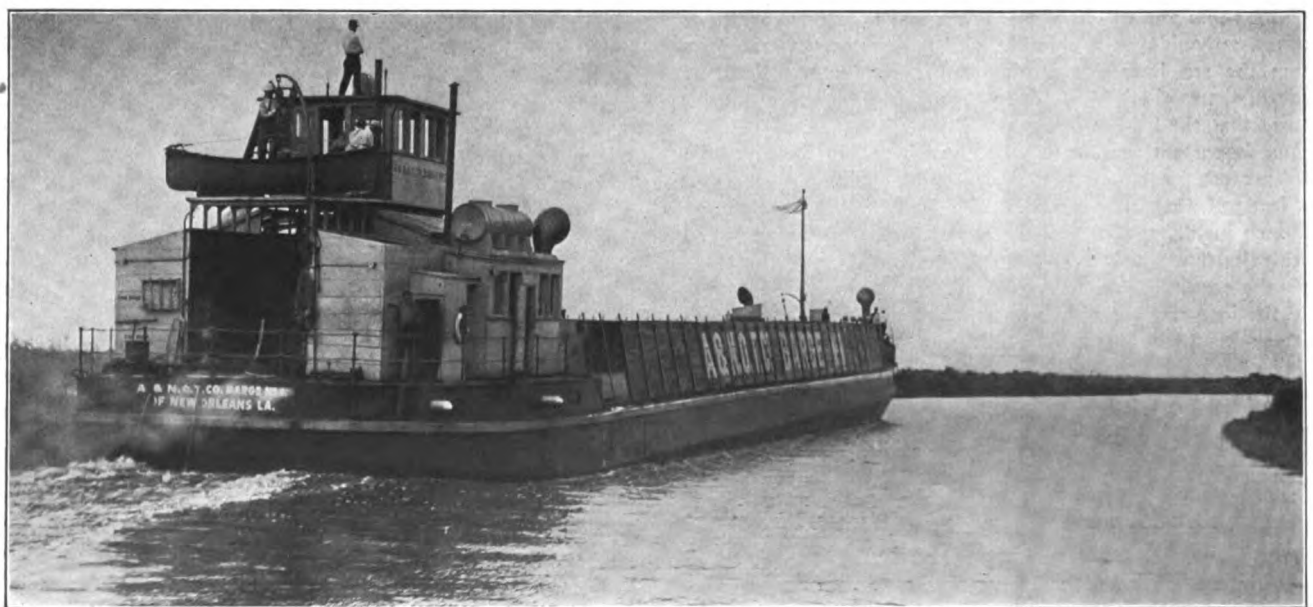


Fig. 4.—Stern View of 1,000-Ton Alabama Coal Barge Entering Lake Borgne Canal at Full Speed

in spite of the fact that the stern-wheeler carries only one-third of the tonnage which the new barges carry. The cost of operation of the stern-wheeler, furthermore, is about four times the cost of operation of the barges.

SELF-PROPELLED BARGES OF THE ALABAMA & NEW ORLEANS
TRANSPORTATION COMPANY

The new barges, a description of which was published on

page 467 of the November, 1913, issue of INTERNATIONAL MARINE ENGINEERING, are so simple in construction that they were built practically in the heart of the Louisiana swamps, 14 miles from New Orleans, and most of the work was done by unskilled labor, consisting principally of negroes from the cane fields. Yet the barges are entirely seaworthy, as shown by the fact that on October 22, 1913, three of them were in the Mississippi Sound during a severe hurricane, when two ocean-

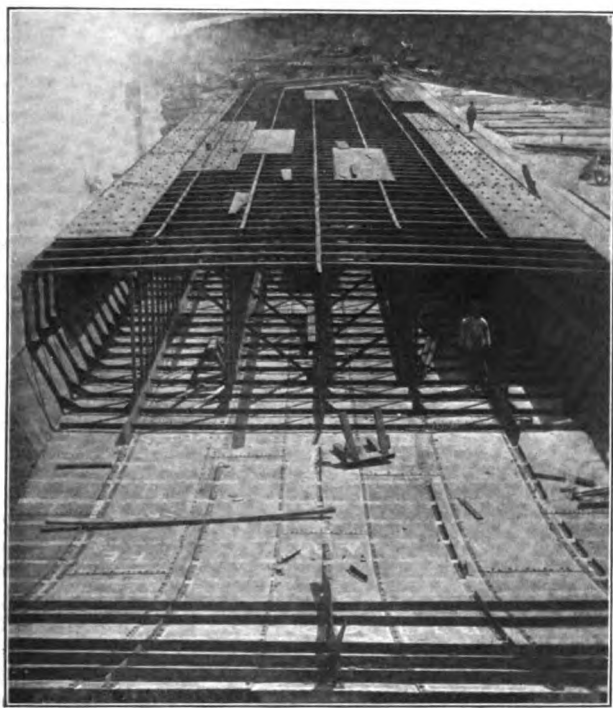


Fig. 5.—Barge Under Construction

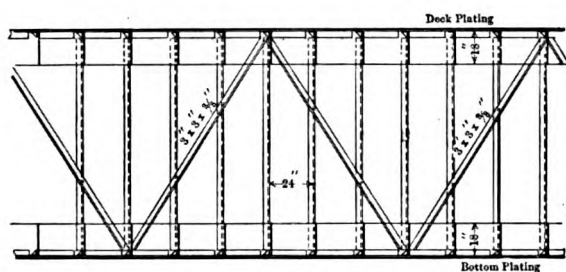


Fig. 6.—Section of Longitudinal Girder

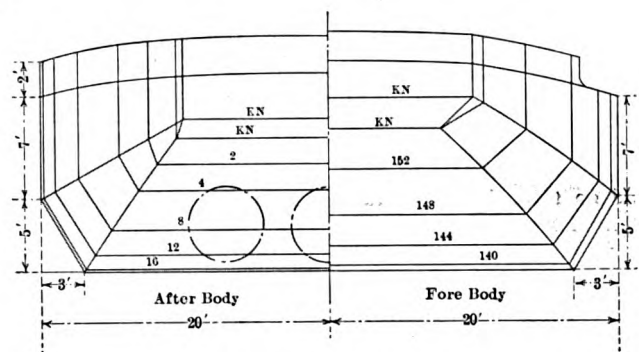


Fig. 8.—Body Plan

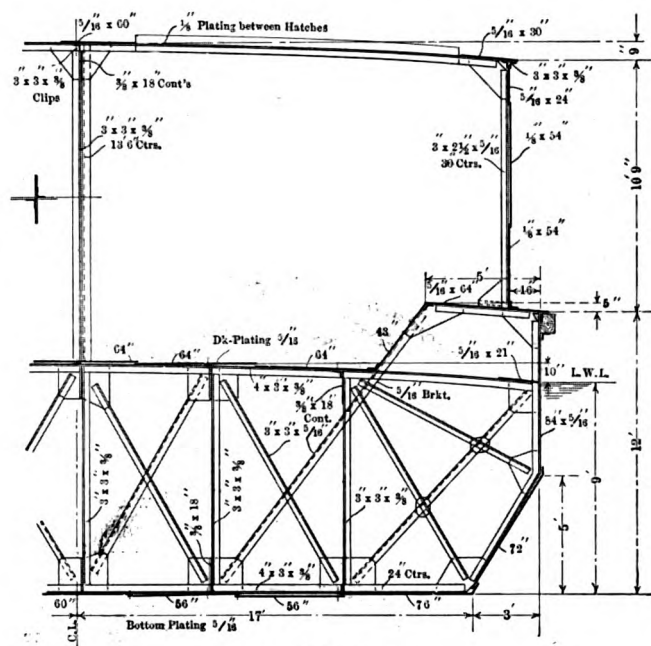


Fig. 9.—Midship Section

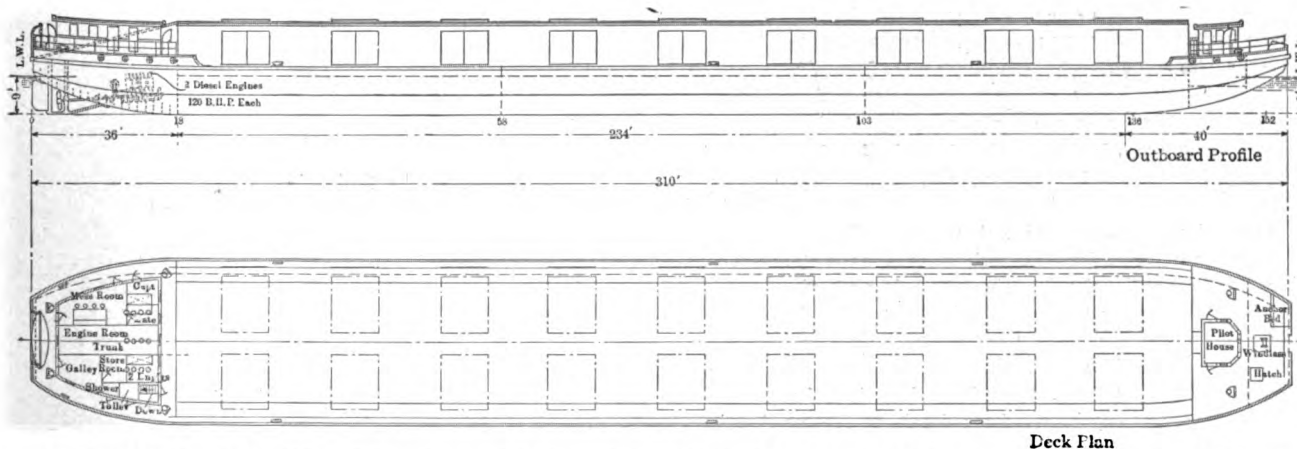


Fig. 7.—Profile and Deck Plan of 2,400-Ton, Triple-Screw Barge Designed for New York State Barge Canal by John H. Bernard

going vessels which were lying in the harbor at Gulfport were damaged, and a Government revenue cutter, which was stationed in the Sound, was obliged to put into Mobile for protection, while the three barges successfully fought out the storm without the least damage to hull or cargo.

The barges are of steel, and are propelled by two 75-horsepower internal-combustion engines, operating on producer gas, which gives them a speed of $7\frac{1}{2}$ miles per hour when empty and $7\frac{1}{2}$ miles per hour when loaded. The fact that the same speed is obtained when the barges are either empty or loaded is due to the fact that when fully loaded the propellers are better submerged, so that what the boat loses in speed through skin friction is gained through the better grip the propellers have on the water. Furthermore, these barges do not "squat." They can enter the narrow channel of Lake Borgne Canal, which, as stated above, is only 80 feet wide and 8 feet deep, and go through this at full speed, with the result that the stern lifts rather than settles. Moreover, the wash of the barge is practically nil.

In Fig. 4 is shown a stern view of one of the barges when moving at full speed in the entrance of Lake Borgne Canal, while in Fig. 1 one of the barges is shown in the open Sound, also at full speed. In both instances the wash is hardly sufficient to rock a rowboat. A barge which produces as little wash as this would undoubtedly mean a saving of millions of dollars to the State of New York in protecting the banks of the canal, and it might also mean an increase in the capacity of the canal of over 50 percent by permitting the barges to run at full speed.

An added advantage to the navigator which this barge has is the fact that it does not swerve. As it is propelled by two propellers, spaced 10 feet apart and steered by three rudders—two wing rudders and one center balanced rudder—it has been found very steady in its movements.

Although built for speed the barge apparently has a clumsy form. This shape, however, has been developed only after years of experiment, and it is a design that has been universally adopted for shallow draft navigation in Holland, more particularly in Amsterdam, a city which is cut up by innumerable canals, all shallow, straight and narrow. Although the Dutch canal boats are designed primarily for canal work, yet the same craft are very often obliged to go, with valuable cargoes, such as rice, sugar, tobacco, etc., into the open harbor of Amsterdam, which is large and wide enough to demand comparatively seaworthy boats.

The barges are very simple in construction, doing away with any special banded plate, their sides having a slant, which, in this case, should be the same as the average slant of the New York Barge Canal. This means that the barge can come closer to shore when receiving or unloading its cargo, and also that the barge can hug the shore closer when going at full speed, which will permit a greater space between similar barges when passing each other in the canal. These barges can turn at full speed over a radius of one and one-half times their own length.

PROPOSED BARGE FOR NEW YORK STATE BARGE CANAL

For the New York Barge Canal a similar barge can be designed, which would be very economical and safe to operate, not only through the entire canal and through the Hudson River or Lake Oneida, but also through the Great Lakes. Of course, certain alterations must be made to meet local conditions; for example, the fact that the bridges limit the clearance above the water to 15 feet 6 inches, makes it necessary that under no conditions shall any part of the barge be more than 15 feet above the water.

Fig. 7 shows the general outline of such a barge. The dimensions are: Length, 310 feet; depth, 12 feet at the sides and 12 feet 6 inches at the center; draft, when fully loaded, 9 feet; width of deck, 40 feet, and width of bottom 37 feet.

The barge is flat-bottomed, and is propelled by three 120-horsepower Diesel engines. Such a barge will have a net carrying capacity of 2,400 tons and a speed of 8 miles per hour when fully loaded. When empty the barge can be quickly loaded by means of ballast water, which can give the boat any desired draft.

The barge is equipped with three rudders; that is, two side rudders and one center balanced rudder. The pilot house is forward, together with quarters for a crew of ten men. The captain, the two mates and two engineers are berthed aft, where the galley is also located. The cargo box is 12 feet high, 36 feet wide and 265 feet long, inside dimensions. The deck is sunken 3 feet, giving an additional height of 3 feet to the cargo box, so that the total space in the cargo box is 140,000 cubic feet. Since the barge draws only 9 feet of water it will have a freeboard of 3 feet. The bow rises 8 feet above the waterline, while the deck over the engine room rises 2 feet over the main deck.

Such a barge, complete with all machinery, electric lights, winches, pumps, bulkheads, fully equipped, will cost f. o. b. New York State Barge Canal \$75,000 (£15,400). Granting for a moment that the canal will be closed six months of the year (which seems to be unreasonable and which will probably be changed to not over four and one-half months, perhaps four months), the cost of operation of the barge figures out, allowing 20 percent for interest, depreciation, insurance and repairs, as a total of \$15,000 (£3,080) for each season.

Figuring the speed of the barge at 8 miles per hour, and allowing for the narrow parts of the canal, where the barge can make only 4 miles per hour, and also allowing ample time for loading and unloading at either end of the canal and at intermediate points, the barge ought to make a round trip every week. The barges running from the Black Warrior coal fields in Alabama to New Orleans, a distance of 515 miles, going through seventeen locks, make the round trip, including loading and unloading, and taking into consideration all delays, such as fog, etc., in an average of nine days. So certainly we are making a liberal allowance for delays when we figure on this barge making only one round trip per week, or twenty-five trips per season.

With the above assumptions the total overhead charges per round trip amount to \$600 (£123).

COST OF OPERATION

The 1,000-ton barges running between Alabama and New Orleans have a total crew of seven men for day and night runs—a captain and mate, two deck hands, two engineers and a cook. But to avoid criticism, let us equip this barge for the New York Barge Canal with a crew twice as large, as follows:

1 captain, at monthly salary of.....	\$200	(41/13/4)
2 mates, at \$100 (20/16/8) each per month.....	200	(41/13/4)
2 quartermasters, at \$60 (12/10/0) each per month.....	120	(25/0/0)
4 deckhands, at \$40 (8/6/8) each per month.....	160	(33/6/8)
2 engineers, at \$100 (20/16/8) each per month.....	200	(41/13/4)
2 oilers, at \$60 (12/10/0) each per month.....	120	(25/0/0)
1 cook, per month.....	40	(8/6/8)
14 men, a total per month of.....	\$1,040	(216/13/4)
or \$250 (52/1/8) per trip.		

Figuring the subsistence at 40 cents (1/8) per man for every day of the trip, gives a total subsistence of $14 \times 40 \times 7 = \$40.00$ (8/0/8). The fuel, when using Diesel engines, is very liberally figured at \$1.44 (30/0/2) per round trip. Granting for each trip for supplies, such as oil, waste, ropes, paint, etc., the liberal sum of \$66 (13/15/0), we have a total outlay for each trip of \$1,100 (£226). Thus for a round trip a barge having a carrying capacity of 2,400 tons will be able to move 4,800 tons for \$1,100 (£226), or slightly less than 23 cents (0/11½) per ton.

Assuming that one company uses thirty barges, such a company could move during the open season 3,600,000 tons of freight at a charge of 26 cents (1/1) per ton, and with such charge could cover the cost of operation, have \$100,000

(£20,500) for administration purposes and agents' fees, and have besides 6 percent interest on the capital invested. As the average railroad charge between New York and Buffalo is \$1.96 (8/2) per ton, this water transportation means a saving per ton of \$1.70 (7/1), or approximately 87 percent.

DOES WATER TRANSPORTATION PAY?

The above estimates have been very conservative, so that if any criticism is offered even stronger figures can be brought forward. It is safe to predict, however, that with efficient methods freight can be transported from Buffalo to New

York at a charge of 25 cents (1/0½) per ton and yield handsome profits at such charge. Three years ago the writer investigated the possibility of freight movement between Lake Champlain and New York, and found that freight could be moved for 22 cents (0/11) per ton even though the barges returned empty.

Although the above figures are based on the assumption that the barges will be idle for six months of the year, that does not mean that even should the canal be closed the barges need be idle. They are sufficiently seaworthy to be sent down the coast to Texas and Louisiana, if so desired, where they can be profitably employed.

The Westinghouse System of Bridge Control

Apparatus for Controlling a Turbine-Driven Ship from the Bridge—The Governing Mechanism, Gages and Speed Indicators

One of the most interesting features of the marine turbine drive developed by the Westinghouse Machine Company, East Pittsburg, Pa., which involves the use of high-speed turbines with a floating frame reduction gear interposed between the turbine and the propeller shaft, is the method by which the turbines can be controlled by the navigator on the bridge as well as by the engineer from the starting platform.

After several months' service on the United States collier *Neptune*, where this system was first installed, the method of bridge control was found to be thoroughly reliable and of distinct advantage in the operation and maneuvering of a ship in an open seaway and in going alongside a dock. Indicators show the navigator on the bridge just what the turbines are set to do and just what they are doing at every instant, while gages show the air and steam pressures available, so that the action of the propellers is always directly under the control of the officer who is handling the vessel.

The medium used for transmitting to the engine room the movements of the control handle on the bridge is air, while the medium used for the actual operation of the relays moving the steam valve is oil.

DETAILS OF THE CONTROL SYSTEM

A diagrammatic arrangement of the operating parts of the control system, including the bridge control and steam nozzle valves, is shown in Fig. 3. As will be seen from examining the left-hand part of the illustration, the bridge control lever *D'* is movable in two quadrants *A'* and *C'*, with an offset *B'* parallel to the axis of movement.

When the control lever *D'* is at *B'* both the ahead and astern nozzles are closed. The operating handle *D'* turns a shaft carrying a cam *E'*, which has pressed against it the point of a small valve *F'*. The valve *F'* is hollow and closed at both ends, and has two sets of ports, *G'* and *H'*, the former

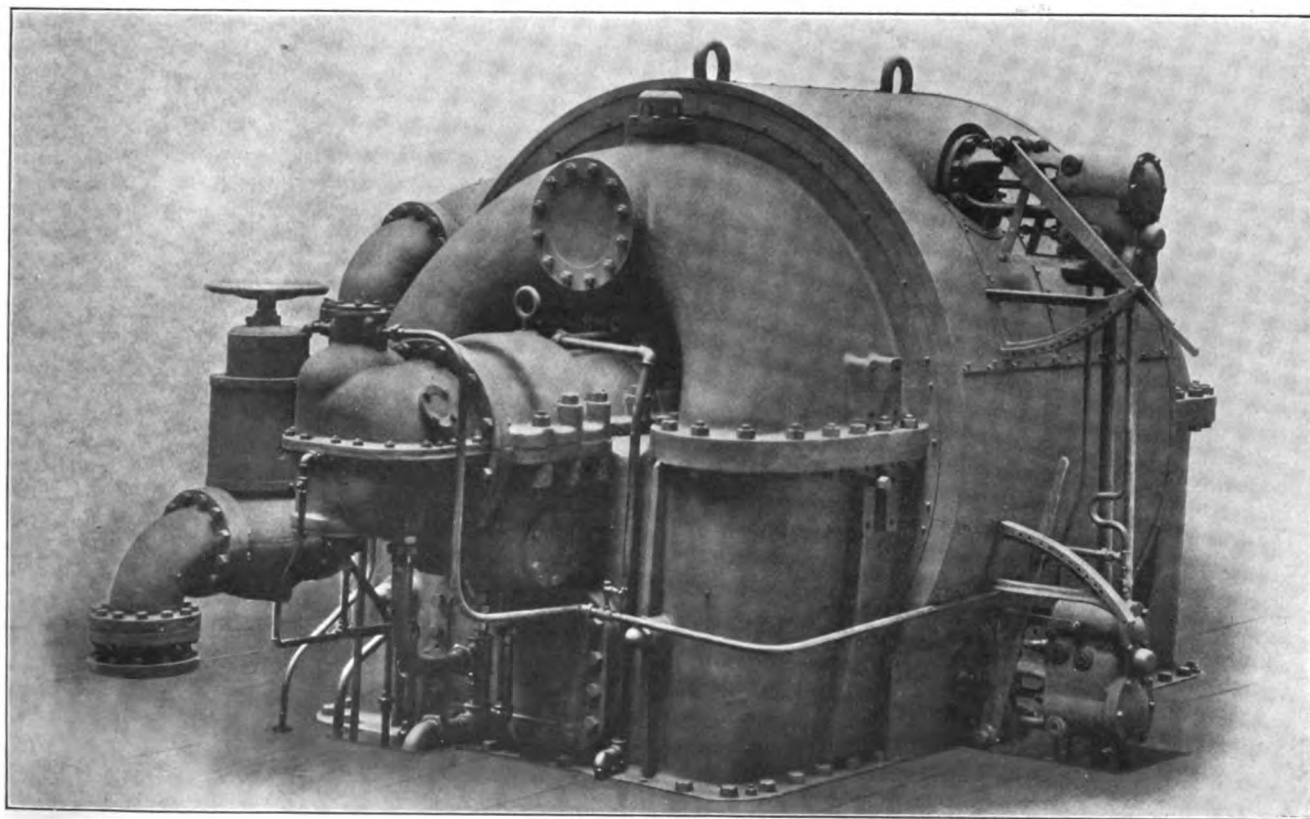


Fig. 1.—Oil Relays for Operating Ahead and Astern Nozzle Valves

always being in communication with the source of high-pressure air supply (furnished by a standard Westinghouse air brake compressor), and the latter having its edge to the extreme left, just line in line with the ports I' in the piston L' . The end of the plunger valve F' is line in line with the left-hand edge of the ports I' . A small spring K' within the piston L' is provided to maintain a positive contact between the point of the plunger F' and the cam E' , thus fixing the position of the plunger F' by its point of contact with the cam E' . The movement of the piston L' to the left is resisted by the spring N' . The space J' , to the right of the piston L' , communicates with space Q' between two diaphragms $R' R'$,

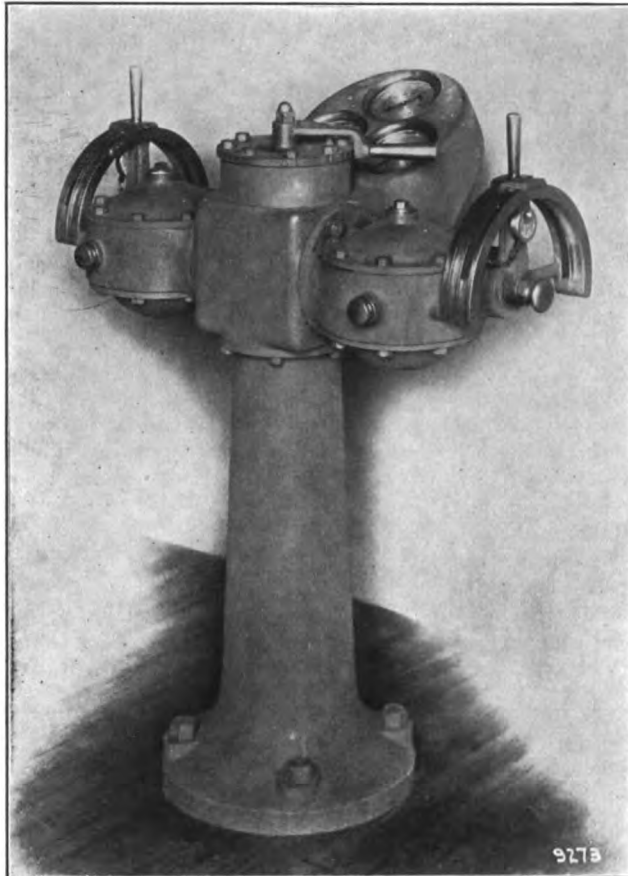


Fig. 2.—Bridge Control Stand

and the space to the left of the piston L' communicates with the atmosphere. A communication to the passage within the piston L' to atmosphere is provided by the ports M' .

The diaphragms $R' R'$ control two valves, T' and U' , the former controlling an opening to atmosphere and the latter an opening to the source of high-pressure air supply. The spaces $S' S'$ are connected by a passage, so that the pressure in them is always equal. The space S' communicates either with the ahead or the astern operating relay in the engine room.

The function of the bridge control valve just described is simply to maintain a predetermined constant pressure in the air relay cylinder, and this is accomplished as follows with the apparatus just described.

OPERATION

When the lever D' is moved from its central position, B' in the ahead quadrant A' , the cam E' pushes the valve F' to the left, thereby bringing the port H' into communication with the port I' in the piston L' . This admits high-pressure air into the space J' , and consequently Q' . As the pressure in the space J' increases, it will quickly force the piston L' to

the left, until the right-hand edge of the port I' cuts off communication with the port H' , or should there be leakage, it will take a position such that the opening through the ports H' and I' would be just large enough to pass sufficient air to make up for the leakage and maintain a constant pressure in the space J' .

Since, normally, the spaces $S' S'$ are at atmospheric pressure, when the pressure between them in the space Q' is increased, the diaphragms are forced outwards, closing the valve T' and opening the valve U' , thus admitting high-pressure air into the space $S' S'$ until the pressure in the latter is equalized with the pressure in the space Q' , thus bringing the diaphragms back to their normal position, permitting the valve U' to seat and thus cut off the supply of high-pressure air. The pressure existing in the space S' is transmitted to the space P to the left of the air relay cylinder in the engine room.

As in the case with the piston L' and the ports I' and H' , should there be leakage in the piping system communicating with the space S' , the diaphragms $R' R'$ will be distended slightly, thus holding the valve U' open sufficiently to make up the leakage and maintain constant pressure. Conversely, it is evident that if the pressure in the space J' should increase above that predetermined for a given position of the piston F' , the piston L' will move to the left, and open communication between the space J' and the inside of the piston L' , thus exhausting air from the space J' to atmosphere through the port M' . This condition also arises when operating lever D' is moved towards the off or central position, which permits the movement of the valve F' to the right. In the latter case the pressure in the space Q' will be decreased and become less than the pressure in the spaces $S' S'$, causing the diaphragms $R' R'$ to collapse and open the valve T' to atmosphere until the pressure in the spaces $S' S'$ is decreased to that in the space Q' .

When the control lever D' is to be moved to the astern position, it is necessary to move it in the direction of its axis of rotation in the slot B' . When this is done an arm W' , held between two collars on the cam shaft, moves the small slide valve X' to the right, and brings the pipe Z' to the astern air relay cylinder into communication with space S' , and the pipe Z'' into communication with the atmospheric exhaust.

From the above it will be seen that it is impossible for the operator to become confused and accidentally move the control lever in the wrong direction, or try to put it in both the ahead and astern position at the same time, as it is necessary to move the controlling lever sideways before it can be moved from one position to the other.

THE GOVERNOR

The governor, which controls the speed of rotation of the turbine shaft, is shown on the right of the diagrammatic sketch, Fig. 3. As will be seen, it is of the common fly-ball type, and is driven through the turbine shaft by a worm E and wheel D . The only movable part of the governor is the hollow spindle B , having the ports N and H in it. The movement of the spindle B is restricted by a diaphragm K , to which it is attached. The upper side of the diaphragm is open to atmosphere, and the space below G is filled with oil.

High-pressure oil at about 65 pounds gage per square inch is admitted to the revolving spindle B through the ports A .

When the governor spindle is revolved by the turbine shaft the balls $M M$ are thrown outward by centrifugal force, thus pulling the spindle B downward against the resistance of the diaphragm K , opening communication through the port H in the spindle, and I in the upper spindle bushing, to the space G , the latter communicating through a port L to a pipe connected to the space N of the air cylinder relay.

As the pressure in the space G , below the diaphragm K , increases, it resists the downward thrust of the spindle B ,

due to the centrifugal action on the governor weights until the upward pressure on the diaphragm exactly balances the downward thrust of the spindle, thus permitting the upward movement of the spindle *B* by the elasticity of the diaphragm *K* until the communication through the ports *H* and *I* is cut off. Thus it is evident that for any given number of revolutions of the governor spindle there must be a constant fixed pressure of equilibrium in the space *G* below the diaphragm. Should the pressure in the space *G* exceed that corresponding to the speed of rotation of the governor at any instant, the spindle *B* will be forced upward, thus opening communication between the space *G* and the atmosphere through the ports *N* in the governor spindle body.

AIR RELAYS

In the center part of the diagrammatic sketch is shown the air relay cylinder, relay valve and operating cylinder, which control the opening and closing of the nozzles of the turbine according to the speed of rotation desired. Referring to this part of the illustration, and starting with the turbine at rest, the piston *Y*, which opens and closes the nozzle valves, is at the farthest end of its travel to the right, and all the turbine nozzles are closed. When air from the bridge control valve is admitted to the space *P*, to the left of the air-operating cylinder, the piston *Q* will be forced to the right against the resistance of a small spring, thus carrying the relay plunger *O* with it. The movement of the latter establishes communication between the space *R*, supplied with high-pressure oil, and the space *X*, to the right of the piston *Y*, and the space *W*, to the left of the piston *Y*, is brought into communication with the space *T*, and thereby with the oil return to the reservoir, which is open to atmosphere. Thus the piston *Y* will be forced to the left by the oil pressure and through a rack engaging with the gear cut in the periphery of the nozzle valve, will rotate the latter and open the high-pressure nozzles, thus admitting steam to the turbine. As there is nothing to resist the motion of the piston *Y*, nozzles in excess of those necessary to give the desired speed are opened instantaneously, and thus the turbine begins to speed up very rapidly. Now, however, as the turbine gains speed and the governor revolves, the governor spindle *B* is forced down and high-pressure oil is admitted through the ports *H* and *I* and through the passage *L* to the space *N* to the right of the piston *Q*, thus tending to push it to the left against the air pressure in the space *P*. As the piston *Q* moves to the left it carries the relay plunger *O* with it, and when the oil pressure in the space *N* balances the air pressure in the space *P*, or, in other words, the space *J'* in the bridge control valve, the piston *Q* will take a fixed position, and the piston *Y* will take up a position such that the ports communicating with *W* and *X* are both closed, thus maintaining a constant speed of revolution.

Should the turbine tend to exceed the speed fixed by the air pressure in the space *P*, the piston *Q* would move to the left and bring the space *X* into communication with the space *S*, connected to the oil exhaust, and the space *W* would be brought into communication with the space *R*, supplied with high-pressure oil. This would cause the piston *Y* to move to the right until sufficient nozzles had been shut off to reduce the speed, until the oil pressure in the space *N* was again brought into equilibrium with the air pressure in the space *P*.

"CUT-OUT" VALVE

The bridge operating stand which supports the bridge control valves is shown in Fig. 2, from which it will be seen that there is a rotary slide valve on top of the stand between the port and starboard control valves. This latter valve is not shown in the diagrammatic sketch previously described, and is not shown in detail because of the difficulty of showing the various ports in it. Its function, however, is to permit operating both turbines, either ahead or astern, by either the port

or starboard control valve, or to cut out both the starboard and port control valves. In addition to these functions which this valve performs on the bridge control stand, the similar valve on the control stand in the engine room has a position for cutting out the bridge stand and other operating stands in other portions of the ship. Thus when the control system on the bridge is connected, all control stands excepting that in the engine room are made inactive, while if the stand in the engine room is being used all other stands (including the one on the bridge) are disconnected. This arrangement avoids interference, permits rapid changes from one station to another, but always leaves the engine room stand operative in case of an emergency.

Ordinarily, except when maneuvering, the bridge stand is always connected up so that both turbines are controlled by moving either the port or starboard control levers, which simplifies the operation and permits controlling both turbines

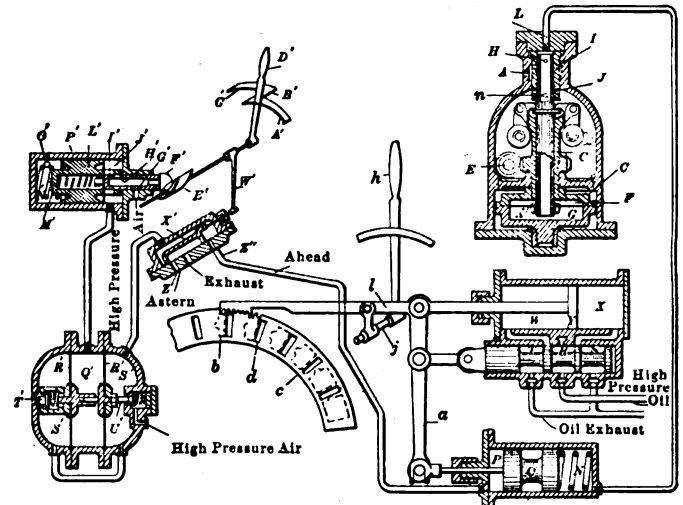


Fig. 3.—Diagrammatic Arrangement of Bridge Control and Nozzle Valves

if either the port or starboard operating valves should for any reason become temporarily disarranged.

In addition to the standard screw-down valves on the ahead and astern turbines, each turbine is provided with a hand-operating device, by means of which a nozzle valve can be moved by hand, if necessary or desirable.

This is accomplished by the lever *h*, as shown in the diagrammatic sketch, Fig. 3. The lever is pivoted on the shaft *j*, and has a lever engaging with the crosshead on the valve-operating piston rod *l*. The latter lever is fitted with a latch so that it can be quickly connected or disconnected when not in use.

As previously pointed out, there may be a number of operating stations from which the turbines can be controlled when this is desirable, as, for instance, there might be considerable advantage in the case of battleships in having the turbines controlled from the central station as well as from the bridge and starting platform.

GAGES AND INDICATORS

In order that the officer on the bridge or navigator can see that the turbines are operating as desired, and also to show that the control system is in operating condition, gages are provided which show the steam pressure in the boilers, the air pressure in the control system, the air pressure in the pipe lines *Z' Z''*, as well as the pressure in the oil supply system, and the pressure of the oil under the pistons of the floating frame on the reduction gears.

As it would be frequently necessary to have the piping communicating from the engine room to the bridge in exposed places where it would be liable to freeze, and because of the

difficulty of allowing for the hydrostatic head in the pipe between the bridge and the engine room in the case of liquids, the steam and oil pressures are indicated on the bridge by means of compressed air and a small relay valve, such as illustrated in Fig. 4. As the steam pressure or oil pressure to be indicated on the bridge may exceed the air pressure available, the former are reduced in some ratio, such as 2 to 1 or 3 to 1, as required.

In Fig. 4 the orifice *B* is connected to the steam or oil line, and the pressure desired is indicated on the bridge by the air pressure in the pipe connection to *F*, which communicates with the gage on the bridge. The operating element of the relay consists of the plunger *L A*, having *L* and *A* the same diameter, or *A* a smaller diameter than *L*, for relaying oil or steam pressures. The plunger *L A* is fitted in a bushing, having ports *G* and *H*, port *G* communicating with the high-pressure air supply and port *H* communicating with atmos-

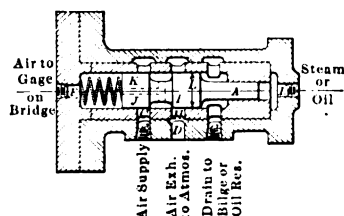


Fig. 4

phere. A drain *C* is also provided, which may either drain to the bilge or to the oil reservoir when oil pressure is being relayed.

Normally, when not being used, the small spring pushes the plunger *L A* to the right, and the gage on the bridge is then connected to atmosphere through the ports *K* and *H*. When steam or oil under pressure is admitted at *B*, the plunger *L A* is forced to the left against the resistance of the spring, and the edge *J* of the plunger *L A* moves over the port *G* and admits high-pressure air through the port *K* to the gage on the bridge, the pressure in the space to the left of the plunger *L A* increases, until the pressure times the area of the left-hand portion of the plunger *L A* equals the pressure times the area on the right-hand portion of the plunger *L A*; thus if the diameter of plunger *L* is twice the diameter *A*, the area of the large end of the plunger will be four times that of the small end, and the actual pressure maintained on the left-hand side of the plunger will be one-fourth of that maintained on the right-hand side of the plunger, thus 200 pounds gage steam pressure would be indicated on the bridge by 50 pounds of air pressure. The oil pressures are transmitted to the bridge in the same way.

SPEED INDICATORS

As was pointed out in describing the operation of the controlling system, there is a fixed pressure (for any given number of revolutions per minute of the turbine shaft) in the spaces *Q'* and *S' S'*, as well as under the diaphragm *K* in the governor spindle and the space *N* to the right of the air-operating cylinder. Consequently, instead of graduating the gage recording the air pressure in the space *S'* in pounds, this scale may conveniently be graduated to read revolutions per minute, so that those in charge on the bridge can see every instant exactly the number of revolutions which the propellers are making.

Another novel feature of the new control apparatus used on the U. S. *Neptune* is a set of speed recording instruments. As previously mentioned, for any given turbine speed there will be a definite oil pressure under the diaphragm *K* of the governor, and consequently this pressure can be used to indicate the speed of the turbine and propeller.

Pressure gages of the recording type may be used, showing the speed, direction of rotation and period of operation of

each turbine under any given conditions. These records furnish an accurate log which is indisputable in case of a controversy regarding the speed or manner in which the turbines were operated. These cards can be graduated to read revolutions or knots, as desired.

The Present Position of the Marine Gasoline (Petrol) Engine

BY L. B. CHAPMAN, A. M. I. N. A.

A great deal has been written about the progress of the automobile engine, and the various phases of its development have been kept constantly before the public. The marine engine, on the other hand, has been given very little attention, and a systematic analysis of it has never been attempted. In this article a few of the characteristics will be taken up and considerable comparative data presented, mostly in the form of curves. This information shows clearly what has been accomplished in this field and some comparisons of the marine and automobile types are presented. The data should be of great value to the naval architect, engine designer and purchaser, and is worthy of considerable thought and study.

As will be shown shortly, the marine type of engine presents a great diversity in designs as compared to the automobile engine. The reasons for this are easy to understand. The automobile engine builder went to work in a serious and scientific manner to perfect the gasoline (petrol) engine, while the marine engine builder went at it by the "cut-and-try" method, adopting the important details from the automobile engine. Let us consider the various curves in detail.

PRICE

Fig. 1 shows curves of prices of twelve representative engines plotted against horsepower. Catalogue prices could not be used direct on account of the wide discounts allowed by

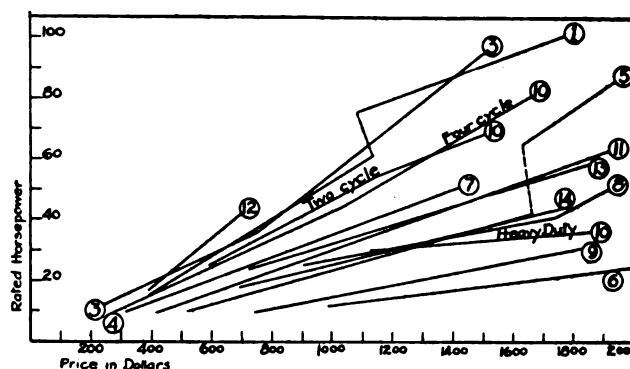


Fig. 1.—Price Includes Engine, Ignition, Etc., Propeller, Gear, and Bosch Magneto

the various manufacturers, but were corrected by the discounts allowed, which were obtained directly from the manufacturer, and in every case include the same equipment. Thus all the engines are reduced to a common basis of comparison. It was found that the prices of each make of engine fell very nearly on a straight line, thus showing that prices vary directly with the horsepower; but each make having a different rate of increase, as shown by the slopes of the lines in Fig. 1. This series of curves shows an almost unbelievable wide variation in prices; thus a 50-horsepower engine, for instance, varies in price anywhere from \$900 to \$1,800 (£183 to £370). Even wider variations than this can be found by an inspection of the curves. It would be hard to explain such wide variations in price, especially in view of the keen competition in this field. No doubt some of the variation is due to over-rating and underrating of the power, but this could only account for part of the wide variation. In view of the fact that

all these engines are well-known makes, and to all appearances of equal merit, the subject of prices is worthy of careful investigation. This curve sheet is presented not for its particular value but simply to show the general trend of prices and the unsettled condition in this field.

WEIGHTS

Fig. 2 shows curves of weights per horsepower of twenty-one makes of engines. It was found upon careful investigation that the weights per horsepower varied with the horsepower and with the revolutions, the high-speed engines, of

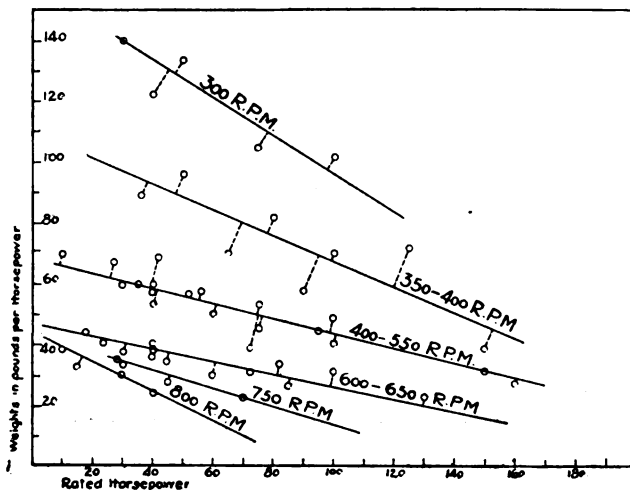


Fig. 2.—Weights of Marine Engines With Reverse Gears; Twenty-one Makes Represented

course, being of a lighter design. Fig. 2 shows the results obtained from the data available, the points representing the various engines falling fairly well on the lines drawn. Thus all the engines selected running between 400 and 550 revolutions per minute fell very nearly on one straight line, the weight per horsepower falling off from 65 pounds to 35 pounds as the power increases.

Fig. 3 shows the weights of high-speed marine engines and covers this field very thoroughly. The results are presented

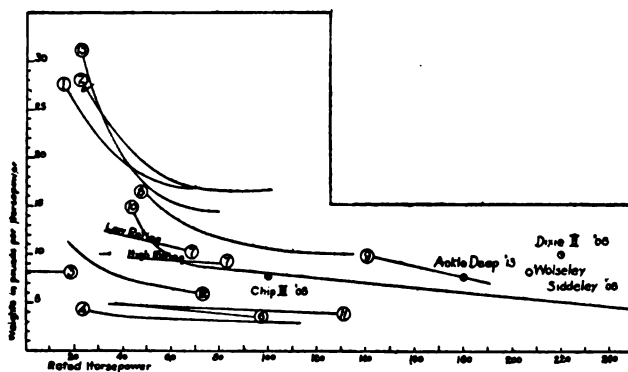


Fig. 3.—Weights of High-Speed Engines: (1) 1,000, Part Aluminum with Gear; (2) 900, Medium Speed with Gear; (3) 950, Canoe Motor; (4) 950 to 1,200, Revolving Aero Motors; (6) 1,000 to 1,600, Vertical Aero Motors; (7) 1,000, High-Speed without Gear; (8) 1,100 to 1,200, High-Speed with Gear; (9) 1,200, High-Speed with Gear; (10) 850 to 1,100, Two-Cycle without Gear; (11) 1,000, Two-Cycle without Gear; (12) 1,000 Two-Cycle without Gear; (13) 1,000, Medium Speed with Gear

a little differently here than in Fig. 2. Here each curve represents a particular make of engine and not a collection of various makes as in the other figure. Thus No. 10, for instance, represents a two-cycle make with revolutions per minute varying between 850 to 1,100, depending upon the horsepower. Another point that should be noticed is that some of the curves are for engines with reverse gears and

some for engines without, and, of course, are not directly comparable. Curves 4 and 6 are for aero engines, and represent the general practice in this field, and give an interesting comparison with the other engines on the sheet. Curves 1, 2 and 13 are semi-speed engines, and are really intermediate between those given in Fig. 2 and the purely high-speed racing engines. A few scattered points are given showing the engine weights of some of the notable racers.

BRAKE MEAN EFFECTIVE PRESSURE

By an analysis of all the available brake-horsepower tests of gasoline (petrol) engines, the marine engine appears to be slightly lower in power than the aero and automobile engines for given bore and piston speed. Thus for four-cycle marine engines the brake mean effective pressure varies from 53 to 83 pounds per square inch, the largest proportions of the tests have a mean effective pressure close to 70. Some engines built especially for racing show higher values than this, the highest reliable value being 91 pounds per square inch. By taking catalogue ratings and solving back for mean effective pressures, about the same run of values are obtained as found by the brake tests.

A great many reliable tests on automobile engines show values of 90 to 100, and values as high as 107 have been obtained. Aero engines, while not, as a rule, as high as this, still surpass the marine engine. No doubt there are some marine engines that have a mean effective pressure as high as those quoted for automobile engines, but the general average appears to be much lower, and values for automobile engines seldom run as low as 55 and 60, which are frequent in marine practice. More data are needed along this line before any conclusion can be drawn, but the data available show up the marine engine in rather a poor light. No doubt this superiority of the automobile engine is due to the scientific methods of design as opposed to the "cut-and-try" method of the average marine engine builder.

Why engines vary so greatly in mean effective pressure, of course, is open to question. Such points as valve areas and lifts, compression, areas of passages, timing of valves and adjustments of the carburetor, are, of course, the main causes, and have not been given enough study by many of the makers. When two engines of the same bore and piston speed show mean effective pressures of 51 and 83, respectively, during a

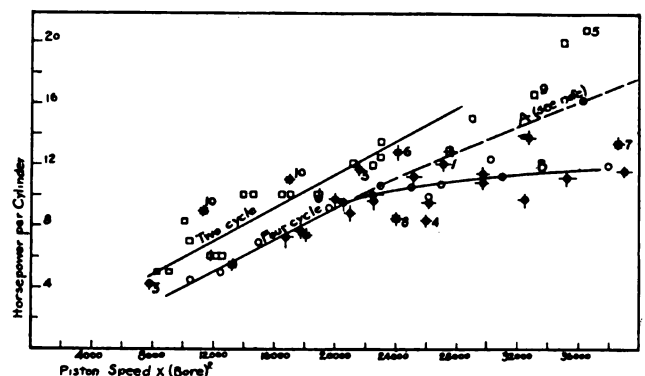


Fig. 4.—A Comparison of Two and Four-Cycle Marine Engines. Squares indicate two-cycle engines as rated (14 makes represented); squares with cross indicate two-cycle brake tests (4 makes represented); circles indicate four-cycle engines as rated (8 makes represented); circles with cross indicate 4 cycle brake tests (14 makes represented). Note A—Dotted line has a number of points beyond the sheet, including brake tests. All points shown are at approximate rated horsepower

competition test, evidently there are still considerable unknown elements in the design. Further information on this subject will be brought out shortly in connection with Figs. 4* and 5. First let us turn our attention to the consideration of the two versus the four-cycle engine.

* The source of much of the data in Fig. 4 is from United States navy tests and the proceedings of the Institution of Automobile Engineers.

TWO VERSUS FOUR CYCLE

From the definition of the cycles the two-cycle engine has twice as many working strokes as a four-cycle engine for a given revolution per minute. Theoretically, then, for the same bore and piston speed the two-cycle engine should develop twice the power that the four-cycle does. This is often the popular opinion, although far from the case, as we shall see shortly. Let us turn our attention to Fig. 4. In these curves the horsepower per cylinder is plotted against piston speed $\times$

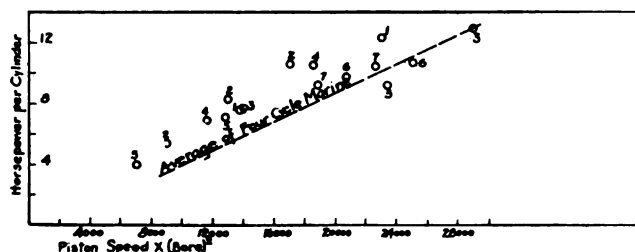


Fig. 5.—Comparison of Marine and Automobile Engines. Circles Indicate Automobile Engine Brake Test

bore² for a large number of engines. Two lines are drawn, one representing the average of the four-cycle points, and one representing the average of the two-cycle points. These curves then give a true comparison of horsepower, and are, of course, proportional to the mean effective pressures. The various points can be distinguished, the squares representing the two-cycle engines and the circles the four-cycle engines. To make the data as complete and valuable as possible, many catalogue ratings as well as all available brake-horsepower tests were plotted. The brake-horsepower tests, all of which are highly reliable, are distinguished from the others by a cross. This figure shows very clearly that the two-cycle engines develop very little more power than the four-cycle for same conditions of piston speed and bore. This, of course, means that the mean effective pressure of a two-cycle engine is little better than one-half that of the four-cycle; that is, the average mean effective pressure is about 35 to 40. Some tests show slightly better than this, but more often they run down in the neighborhood of 30. The cause of this is, of course, due to the poor scavenging of the two-cycle engine. As the scavenging in marine engines is done by the new charge, we would naturally expect an excessive fuel consumption in this type. This, however, is not the case, as the two-cycle engine shows as good, and often better, fuel consumption than the four-cycle engine.

The makers have purposely lowered the mean effective pressure of the two-cycle engines by using smaller ports and a smaller fuel charge in order to obtain good fuel economy. It would thus seem that the two-cycle engine has no place except in small one and two-cylinder engines, where more uniform torque is desired, and in order to reduce the valve mechanism and simplify the moving parts.

Fig. 4 brings out another interesting feature worthy of considerable thought; that is, the falling off of the curve marked *B* above 10-horsepower per cylinder. This is no doubt caused by trying to make a given cylinder develop more power than it is able to; that is, the engines are overrated and are forced in order to produce their power. There are a number of points (some off the sheet as mentioned in note A) that fall on the dotted continuation of the curve marked *A*. These engines, although comparatively few, are no doubt better designed and more fitted to deliver their rated power than those on *B*. The mean effective pressures can be found from these curves by dividing the ordinate reading by the corresponding abscissa and multiplying by 16,800 for four-cycle engines and by 8,400 for two-cycle engines. These two curves not only form a basis for comparison of existing engines but should furnish data for design work.

Fig. 5 shows data of the same nature for automobile engines. The four-cycle marine engine curve is transferred from Fig. 4 for comparison. All the points shown are from brake tests. These points all fall above the marine engine curve, as would be expected from what has been said previously regarding mean effective pressures. If a curve were drawn for these points it would fall off at its upper end the same as *B* in Fig. 4. In this case it is due directly to driving the engine at too great a power. The upper points in every case are for the same engines as shown lower down on the curve, but driven at higher revolutions per minute.

FUEL CONSUMPTION

A point of extreme importance in all engines, of course, is fuel consumption, and although reliable data are scarce, and no curves can be drawn, a few words should be said at this point regarding it. Fuel consumption tests of sixteen marine engines, all using the same quality of gasoline (petrol), showed a range of 0.8 to 1.5 pints per horsepower per hour, with the average figure about 1.3 pints. These engines were all on a competitive test, and, of course, were supposed to be adjusted and in A-1 order. From an exhaustive study of fuel consumption tests by the writer and others, no definite cause can be found for the wide variations in fuel consumptions. No doubt some of it is due to the quality of fuel and adjustments, but probably most of it is due to poor designing. Automobile engines on the test block invariably show better

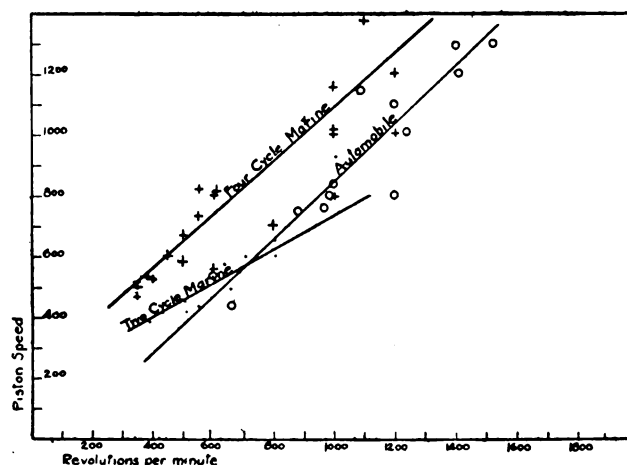


Fig. 6.—Circles Indicate Automobile Engines. Crosses Indicate Four-Cycle Marine, and Dots Indicate Two-Cycle Marine Engines

results than this, values as low as .80 pint per horsepower are common, and figures as low as .65 are not infrequent with a good grade of fuel.

As mentioned earlier the two-cycle engine shows practically no difference from the four-cycle except in engines of extremely poor design.

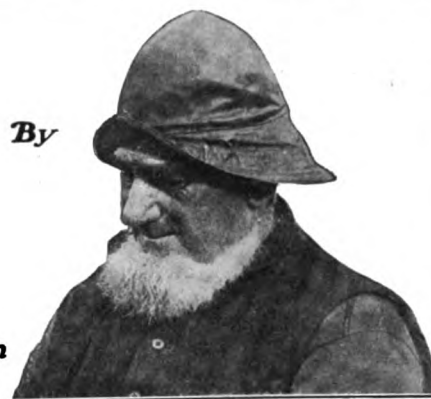
PISTON SPEED

Fig. 6 shows in the form of a curve the present practice in piston speeds both in marine and automobile engines. Contrary to common belief, the marine engine shows a higher piston speed than the automobile engine for the same revolutions per minute. A curve showing the current practice in two-cycle work is also included.

In conclusion, it appears that the automobile engine is slightly further advanced than the marine engine. Little has been said about the aero engine, but in weight, fuel consumption, mean effective pressure and refinement of design, it also exceeds the marine type. The writer has endeavored to bring out a few of the important points, but many important considerations, such as valve arrangements, sparking mechanism and cylinder arrangement, etc., in which it is known that the automobile and aero type exceed the marine type, have been left out.

Economy Talks *By*

"Old Scotch"



A Few Facts About the Chemistry of Combustion

Well, brother marine engineers, the editor of this magazine has given me a license to go ahead with the line of "dope" I handed out to you in the last number, so I am going to devote a few hours each month for some time to come in making good my statement that you can all save money for yourselves, as well as for the owners, by exercising a little more attention to business here and there.

Of course, the boilers are the chief villains in this play, and, as I was telling you last month, it is here that we must look for the escape of the little heat units from the coal we shovel in the furnaces, and see how they escape and what we can do to prevent some of them from getting away at this point.

The elements in the coal which we shovel in the furnace doors are principally carbon and hydrogen so when you hear highbrows talking about hydrocarbons, they simply mean a gas mixture containing those two elements. While the carbon and hydrogen cost money when we buy them in the form of coal, the other necessary element—oxygen—comes free and in abundant quantities with the air. You know this oxygen is really about the most important old gas in the business, and with furnaces the proper regulation of the oxygen as it enters with the air is the principal thing we have to look after in our scramble to burn coal properly and save that 10 percent I am telling you about.

Air contains about 23 percent of oxygen, the remaining 77 percent being nitrogen. We'll forget the latter, as it doesn't figure in the combustion very seriously, anyhow. You may not think that air weighs anything; it does, just the same, but it takes $12\frac{1}{2}$ cubic feet to weigh one pound. To have enough oxygen to combine properly with the carbon and hydrogen in the coal requires on the average about 235 cubic feet of air let into the furnaces for every pound of coal burned. As I said before, it's a good thing we don't have to buy the air at the same rate we pay for coal. When the coal is first thrown on the fire it does not burn immediately, but is baked, we might say, the same as they bake it in a gas retort, and the carbon and hydrogen are driven off in the form of gases. As these same gases arise in the furnaces they meet with the air, which has come up through the grates, and if everything is agreeable for the wedding they unite with the oxygen in the air, the hydrogen going first, as Old Mr. Oxygen seems to prefer that young lady, and the carbon gets attended to next. This union of these elements must be accomplished under very pleasant conditions to all three parties. There must be just the right amount of air and it must be at just the right temperature. You might think from the fuss made it was an old maid drinking tea. If there is not enough air to go around, the carbon suffers and is liable to drift away, either in the form of smoke or else by attaching itself to the fire surfaces in the form of soot. If the air is not sufficiently warmed up the hydrogen and carbon only half-heartedly unite and much of the good effect of the union is lost in the form of water vapor. On the other hand, should the hydrogen and carbon liberated from the coal be excessively heated, they do not unite freely with

the oxygen in the air and certain portions of them pass up the stack in their original form.

The formation of smoke indicates imperfect combustion, as the black clouds passing out the funnels are largely composed of small particles of unconsumed carbon. This waste should, of course, be prevented as much as possible, although in smoke itself the loss is rather immaterial. It is the bad condition of the combustion, indicated by the smoke, which causes the greater losses. The burning of the hydrocarbons shows clearly that air must be let into the furnaces and combustion chambers above the fire as well as beneath it. After the hydrocarbons are liberated, what remains of the coal is in reality like the coke we buy from the gas companies, and it consists almost entirely of pure carbon. This carbon unites with the oxygen forced by the draft up through the burning coal. The oxygen in the air unites with this carbon in what might be termed a natural proportion; that is, one atom of carbon grabs hold of two atoms of the oxygen and forms what is known to the chemical trade as carbondioxide, or carbonic acid gas; to ordinary fellows, like us marine engineers, it is the same stuff that makes soda water sizzle, and produces that flare-back in your nose when you are foolish enough to drink soda water.

In this proportion one pound of carbon absorbs two and two-third pounds of oxygen, and that is the real thing in combustion, or combustion according to Hoyle. But—the pesky stuff gets gay as it reaches the colder layer of fuel on top of the fire and absorbs more carbon, just as a West street longshoreman absorbs schooners of suds on a hot day, and the result is a new mixture in the proportion of one pound of carbon to one and one-third pounds of oxygen. This is the same "pizen" stuff that we smell as coal gas around an anthracite stove when we are huddled up around it in a poorly ventilated room in the winter. Highbrows call this mixture carbon monoxide, and it is about as bad for combustion as the coal gas is for your lungs. If this stuff escaped unburned there would be a big loss of heat units right there, but fortunately when it breaks through the burning coal and runs into more air an additional amount of oxygen is absorbed, so that it is transformed into carbon dioxide, or CO_2 , as the chemists call it, and then real combustion takes place. This illustrates again the necessity of having air put in the furnaces above the grates.

Now, perhaps you do not think all this line of talk I have given you amounts to anything, but believe me, a knowledge of this little chemistry of combustion is a very important proposition, and I can name two ships right now where a careful study of these conditions and the proper regulation of the three elements produced a saving of 10 percent in the fuel. I am going to show just how it was done later on, and if any of you don't believe it can be done, sent a five-spot and I will cover it with an equal amount of my hard-earned wages.

Yours for economy,

Old Scotch

The Watertight Subdivision of Ships and the Effect of Bilging—IV

BY A. L. AYRE

CHANGE OF TRIM DUE TO BILGING

This is an extremely important question, especially in the case of bilging of end compartments, as it may frequently happen that the deck at the end of the vessel will become level with the water, as shown in Fig. 8, or perhaps submerged, and the vessel rendered unseaworthy. The question of trim may be considered in the following form:

In Fig. 13 it is seen that owing to the loss of buoyancy in way of the compartment $x y X Y$, the mean sinkage produced the waterline $W_1 L_1$, but at this waterline the center of buoyancy has moved aft to the position B_1 owing to the withdrawal of the buoyancy at the fore end and its addition in the layer contained between $W L$ and $W_1 L_1$. The horizontal distance

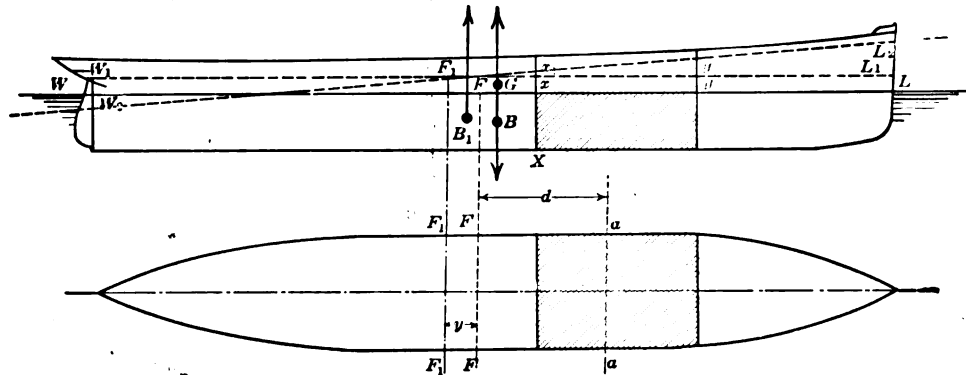


Fig. 13

$B B_1$ represents a couple tending to trim the vessel in a forward direction, and its value can be found as follows:

Let v = the volume of either lost buoyancy or the added layer.

d = the horizontal distance from the center of gravity of the buoyancy contained in the bilged compartment to the position of the center of gravity of the buoyancy contained in the added layer.

And V = the volume of displacement.

Then

$$B B_1 = \frac{v \times d}{V}.$$

This couple multiplied by the total weight of the vessel will give the trimming moment. If the lost buoyancy is taken in tons, then the trimming moment can also be found by multiplying its amount by the distance d . Having found the trimming moment, this can be divided by the moment to change trim 1 inch ($M. C. T. 1''$) and the total amount of trim obtained.

$$M. C. T. 1'' = \frac{W \times G M_2}{L \times 12}.$$

Where W = total weight or displacement in tons.

$G M_2$ = longitudinal metacentric height.

L = length of vessel in feet.

It will be seen that for the new condition $G M_2$ will have altered, on account of the new shape of the displacement and the intact waterplane on both of which it is dependent, seeing that the height of M_2 is determined by longitudinal moment of inertia about the center of flotation, divided by the volume of displacement = $B M_2$, i. e., $I_2 \div V = B M_2$. The value of I_2 will have been greatly affected owing to the dif-

ference in shape and area of the waterplane $W_1 L_1$, as compared with $W L$, while the position of B will have altered in an upward direction. The moment of inertia of the waterplane $W_1 L_1$ will be found by first of all calculating the moment of inertia about the center of flotation ($F_1 F_2$) of same, without having taken into account the lost area in way of the bilged compartment, and corrections afterwards made for the damaged portion as follows:

I_a (the moment of inertia of the portion $x y$ about $F F$) = $I_a + A d^2$, where I_a is the moment of inertia of the $x y$ portion about its axis $a a$; A is the area of the $x y$ portion, and d is the longitudinal distance between $F F$ and $a a$. Then the moment of inertia of the complete waterplane, taken about the axis $F F$, being reduced by the amount of I_a , gives the moment of inertia of the damaged waterplane about $F F$. A further correction is, however, necessary because the axis $F F$ is one for a complete waterplane, while the moment of inertia is required for the waterplane in the damaged con-

dition. We must next, therefore, find the center of flotation of the damaged waterplane, this being shown by $F_1 F_1$ in Fig. 13, the distance aft of $F F$ being y . This second correction, which is also a deduction, is equal to the area of the intact waterplane $\times y^2$, and the actual longitudinal moment of inertia of the damaged waterplane $W_1 L_1$ is then obtained about its own center of flotation. Dividing this corrected moment of inertia by the volume of displacement we obtain the new value of $B_1 M_1$.

The height of the new center of buoyancy, B_1 , for the damaged condition is now found, and this distance, measured above the base, being added to the $B_1 M_1$, gives the height of M above base. If we now deduct from this the height of the center of gravity above the base, we will then obtain the value of $G M_2$ in the damaged condition, and we can therefore use this to obtain the corresponding $M. C. T. 1''$. From this we can then ascertain the amount of trim as previously mentioned, care being taken to distribute the amount at each end in accordance with the position of the center of flotation of the damaged waterplane; for instance, if we assume the length of vessel as being 300 feet, and the new center of flotation at 100 feet forward of the after perpendicular, then 90 inches of trim would be distributed as follows:

$$90 \times \frac{100}{300} = 30 \text{ inches of trim aft,}$$

and

$$90 \times \frac{200}{300} = 60 \text{ inches of trim forward.}$$

It will be noticed that the bilging causes a reduction in longitudinal metacentric height, and this being used in the numerator of the formula giving $M. C. T. 1''$, thereby results in a small value for the latter being obtained, which has the further effect of increasing the amount of resultant trim.

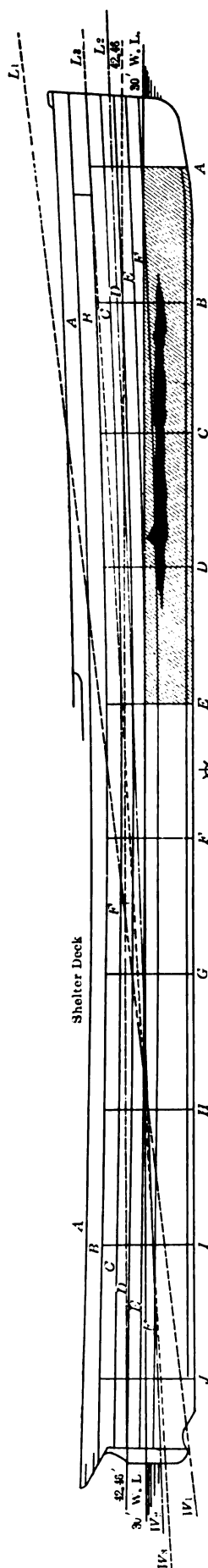


Fig. 14

Had the vessel been fitted with a watertight lower deck at a short distance above the waterline $W L$, so that the amount of flooding would have been limited in a vertical direction, there would have been, in addition to a restriction of the amount of lost buoyancy, a much less serious effect on the trim, owing to the inclusion of such a deck, enabling the original waterplane area to be retained, and in such a case the moment of inertia would have been preserved and a smaller amount of trim resulted, while on account of there being no great shift in the position of the center of flotation the smaller amount of trim would have been more evenly distributed at each end. The watertight deck is therefore seen to have great value from the point of view of trim also.

The following case, which is that of the vessel represented in Fig. 8, shows the effect on the trim due to flooding, resulting from bow damage, causing the loss of buoyancy in the forepeak and No. 1 hold. The dimensions of the vessel are 440 feet length between perpendiculars by 55 feet beam by 33 feet depth, plus shelter deck, the draft being 28 feet.

The amount of lost buoyancy after making allowance for the cargo contained in the space is 1,770 tons, and the area of the intact waterplane, also making allowance for the cargo, is 16,550 square feet. The amount of mean sinkage is therefore

$$\frac{1,770 \times 35}{16,550} = 3.75 \text{ feet.}$$

The total area of waterplane was originally 19,330 square feet and its center of gravity at amidships, and the lost amount of area is 2,780 square feet, having a center of gravity 161 feet forward of amidships.

The center of gravity of the new waterplane is therefore situated at

$$\frac{2,780 \times 161}{19,330 - 2,780} = 27 \text{ feet aft of 'midships;}$$

and this may be taken as also being the position of the center of buoyancy of the added layer.

The center of gravity of the lost buoyancy is 157 feet forward of 'midships. $157 + 27 = 184$ feet, which is the distance between the center of gravity of the lost buoyancy space and the center of the added layer of buoyancy. The trimming moment will be $1,770 \times 184 = 325,680$ foot-tons.

The $M. C. T. 1''$ in the damaged condition is 851, and the total amount of trim is therefore $325,680 \div 851 = 382.7$ inches = 31.9 feet.

By taking into account the sinkage due to the lost buoyancy, and the distribution of trim in accordance with the position of the center of flotation, the new mean drafts are obtained as follows:

	Aft Feet	Forward Feet
Original drafts	28.00	28.00
Mean sinkage	+ 3.75	+ 3.75
Trim aft: $31.9 \times \frac{220 - 27}{440}$ (rise)	- 14.00	
Trim forward: $31.9 \times \frac{220 + 27}{440}$ (fall)		+ 17.90
New drafts in damaged condition:	17.75	49.65

The total depth of this vessel at the stem is 48 feet 8 inches, which shows the shelter deck at the fore end to be awash to the extent of 1 foot; but it must be remembered that although the loss of buoyancy was not arrested until the upper deck reached the water level, there will be an amount of reserve buoyancy preserved intact above this, provided the upper deck is perfectly watertight. This would reduce the amount of trim, but in any case the shelter deck would almost be

submerged, and so dangerously near to the water level as to make the vessel unseaworthy, while she would also be extremely difficult to navigate. The close proximity of the deck to the water level may result in damage to deck fittings, hatches, etc., by means of seas breaking aboard, and such damage would no doubt destroy the watertightness of the shelter deck at the fore end and result in the 'tween decks being flooded. Should this happen, the best possible condition that the vessel could be expected to float in would be that shown by Fig. 8, *i. e.*, with the fore end of the shelter deck submerged; but great danger would certainly exist owing to the existence of the water in the 'tween decks, and which may find its way into some of the other hold compartments or bunkers, etc., and thereby extend the flooding with increasing dangerous effect.

As a concluding example it will be interesting to take the most serious case of flooding resulting from a large vessel coming in contact with submerged ice or wreckage, with such force as to strip off and damage the shell plating on one side from the bow to a position so far aft as to lay open, say, four compartments, or 40 percent of the vessel's length to the sea, and to consider the effect on buoyancy and trim, together with the differences in effect by means of fitting (a) center line longitudinal bulkheads; (b) watertight lower deck. The vessel is represented by Fig. 14, her dimensions being as follows: 800 feet length between perpendiculars by 92 feet beam by 64 feet depth molded to shelter deck by 30 feet average working draft. Including the shell a displacement of 40,500 tons is given by a block coefficient of .64. The displacement contained in way of the bilged compartments, *i. e.*, between bulkheads *A* and *E*, is 17,020 tons, and the center of gravity of same is 167 feet forward of 'midships.

The total area of waterplane at 30 feet draft is 49,280 square feet, and the area of the lost portion is 20,820. The

We may now look into the question of trim, and in connection with this it may be as well to first of all find the *M. C. T. 1"* for the damaged condition, which will, of course, be a figure very much different to that for the normal condition. For this purpose we will deal with the waterplane at 42.46 feet draft, and assuming the vessel to be intact for the time being a "longitudinal metacenter" calculation gives the following results:

Waterplane area, 52,850 square feet.

Center of flotation aft of 'midships, 7.9 feet.

Longitudinal moment of inertia, 1,780,709,000.

Longitudinal *B M* (metacenter above buoyancy), 1,259 feet.

The portion of this new waterplane, in way of the damage, has a total area of 21,275 square feet, *i. e.*, 15,956 square feet lost and 5,319 square feet remaining, and a center of gravity 169.8 feet forward of 'midships.

The amount of lost area is found by calculation to represent an amount of longitudinal moment of inertia about its own axis, *i. e.*, *I_a*, equal to 110,522,500.

This, together with the further correction for *A d²*, which is found to equal 503,000,000, gives the total amount of correction to be made to the longitudinal moment of inertia, as previously found for an intact waterplane at 42.46 feet draft, and having a center of flotation at 7.9 feet aft of 'midships.

Longitudinal moment of inertia of intact water-

plane 1,780,709,000

Deduct (*I_a* + *A d²*) = 110,522,500 + 503,000,000. 613,522,500

I² about center of flotation 7.9' aft 'midships = 1,167,186,500

A further correction is, however, yet necessary so as to obtain the moment of inertia of the damaged waterplane about its actual center of flotation, the latter being first of all found as follows:

CALCULATION FOR DETERMINING THE ACTUAL CENTER OF FLOTATION

Complete waterplane at 42.46 feet draft = 52,850 square feet with center of flotation 7.9 feet aft

Lost area = 15,956 square feet with center of flotation 169.8 feet forward

Area remaining = 36,894 square ft. distance between centers = 177.7 feet

$$\text{Then } \frac{15,956 \times 177.7}{36,894} = 76.9 \text{ feet shift of center of flotation further aft}$$

$$+ 7.9 \text{ feet aft of 'midships at present}$$

The actual C. F. for the damaged condition is therefore 84.8 feet aft of 'midships

center of gravity of the lost portion is 169 feet forward of 'midships.

We must now make the allowances for buoyancy contained in the cargo, bunkers, structure, etc. Taking the amount of lost buoyancy as being 75 percent throughout the damaged compartment, then $17,020 \times .75 = 12,765$ tons actual amount lost. The net amount of waterplane area lost will be $20,820 \times .75 = 15,615$ square feet. The area of the remaining intact waterplane will therefore be $49,280 - 15,615 = 33,665$ square feet. The amount of mean sinkage can now be found:

$$\frac{12,765 \times 35}{33,665} = 13.27 \text{ feet, depth of layer of sinkage.}$$

The intact waterplane area at half depth of this layer is 35,860 square feet, therefore the correct mean sinkage will be very near to

$$\frac{12,765 \times 35}{35,860} = 12.46 \text{ feet, corrected mean sinkage.}$$

$30 + 12.46 = 42.46$ feet mean draft in damaged condition. This mean waterline is shown dotted in Fig. 14.

The distance from the actual center of flotation to the center of flotation about which the moment of inertia has been calculated is 76.9 feet, and as the area of intact waterplane is 36,894 square feet, the final correction is therefore $36,894 \times 76.9^2 = 218,000,000$.

Then

$1,167,186,500 =$ the *M. I.* about center of flotation at 7.9 feet aft of 'midships.

Less 218,000,000

$949,186,500 =$ the corrected longitudinal moment of inertia of the damaged waterplane at 42.46 feet draft.

$\frac{I}{Y} = \frac{949,186,500}{40,500 \times 35} = 669$ feet *B. M.*

Height of *B* above base..... = 23.7

Longitudinal metacenter above base..... 692.7

Center of gravity above base..... 38.2

Longitudinal metacentric height in damaged condition 654.5

The *M. C. T. 1"* for the damaged condition can now be found as follows:

$$M. C. T. 1" = \frac{W \times G M}{L \times 12} = \frac{40,500 \times 654.5}{800 \times 12} = 2,761 \text{ feet.}$$

The trimming moment can now be estimated, and this will be equal to the moment of transference of buoyancy, *i. e.*, the amount of lost buoyancy multiplied by the distance from its center to the center of gravity of the added layer. We have already seen that the net amount of lost buoyancy is 12,765 tons, and that its center is 167 feet forward of 'midships. To obtain the position of the center of gravity of the added layer, we can safely take the position of the center of a waterplane at half depth of same or a mean between the position of the centers of the 30-foot normal waterplane and the 42.46-foot damaged waterplane. For the 42.46-foot waterplane we have just seen that this is situated at 84.8 feet aft of 'midships, while that for the 30-foot waterplane is given by the following:

$$\frac{15,615 \times 169}{33,665} = 78.5 \text{ feet aft of 'midships.}$$

The mean distance is therefore 81.65 feet aft of 'midships, which can be taken as the position of the center of buoyancy of the layer.

We therefore see that 12,765 tons of buoyancy have been transferred from 167 feet forward of 'midships to a position 81.65 feet aft of 'midships, a total shift of 248.65 feet.

The trimming moment is therefore $12,765 \times 248.65 = 3,173,000$ foot-tons, and this divided by the *M. C. T. 1"* gives the total amount of trim as follows: $3,173,000 \div 2,761 = 1,149$ inches of trim. This can now be distributed at either end in accordance with the position of the tipping center, *i. e.*, the center of flotation of the 42.46-foot waterplane, this being at 84.8 feet aft of 'midships.

$$\frac{1,151}{12} \times \frac{315.2}{800} = 37.9 \text{ feet, trim aft.}$$

$$\frac{1,151}{12} \times \frac{484.8}{800} = 58.1 \text{ feet, trim forward.}$$

These amounts of trim produce a waterline as shown by *W, L*, in Fig. 14, where it will be seen to work out at a position very much higher than the extreme fore end of the vessel. Seeing that this metacentric method of calculation presumes the wedge of immersion to counterbalance the wedge of emersion, and that this does not obtain in the present instance, owing to the great difference between the vessel's deck and the new waterline at the fore end, it is obvious that even if the shelter 'tween decks had been taken as perfectly watertight, there could not have been a counterbalancing effect in the wedges, and that the vessel would eventually sink by the head after the lower holds had filled. The sinking would not be rapid at first owing to the time required for the inflow of water to pass from deck to deck, but after the immersion of the shelter deck occurred the sinking of the fore end would be greatly accelerated, as the wedge of immersion now adds very little as the change of trim continues. As has been previously seen a very serious stress effect may cause the vessel to break into two parts now that the buoyant support of the fore end is almost lost.

(To be concluded.)

MONTHLY SHIPBUILDING RETURNS.—The Bureau of Navigation reports 165 sailing, steam and unrigged vessels built in the United States during the month of June. Twelve of these vessels were steel steamships aggregating 26,191 gross tons.

Launch of the Battleship Nevada

The United States battleship *Nevada*, authorized by Congress in 1911, was launched July 11 at the yards of the Fore River Shipbuilding Corporation, Quincy, Mass., and christened by Miss Eleanor Ann Siebert, niece of Governor Oddie, of Nevada. The *Nevada* is a sister ship of the *Oklahoma*, which was launched by the New York Shipbuilding Company, Camden, N. J., March 23.

The *Nevada* is 583 feet long over all, 95 feet $2\frac{3}{4}$ inches beam, and on a mean draft of 28 feet 6 inches displaces 27,500 tons. The vessel will be propelled by Curtis turbines of approximately 24,800 horsepower, actuating twin screws and supplied with steam by Yarrow type boilers burning oil fuel exclusively. The designed speed is 20.5 knots.

The main armament consists of ten 14-inch guns, firing a projectile weighing 1,400 pounds at a muzzle energy of about 66,000 foot-tons. The guns are placed in four turrets, two turrets having two guns each and the other two turrets three guns each.

The main armor belt, which extends 400 feet along each side of the vessel, is $13\frac{1}{2}$ inches thick and $17\frac{1}{2}$ feet wide, extending from 9 feet above to $8\frac{1}{2}$ feet below the water when the vessel is loaded to her mean draft. The three gun turrets are protected by 18-inch armor on the face and 9 and 10-inch armor on the sides, while the roofs are 5 inches thick. The face of the two gun turrets is 16 inches thick and the barbettes are protected by 13-inch armor.

Excellent protection against plunging fire and also against fragments of shells which might be exploded in passing through the ship's plating in wake of the gun deck, is provided by two protective decks, the lower one being $1\frac{1}{2}$ inches thick on the flat and 2 inches on the slope, and the gun deck, which is the upper protective deck, 3 inches thick.

AMERICAN SHIPBUILDING FOR THE YEAR ENDED JUNE 30, 1914.—During the fiscal year ended June 30, 1914, there were built in the United States 1,291 vessels of 311,578 gross tons, compared with 1,648 vessels of 382,304 gross tons for the same period of 1913. One hundred and four steel steamships, aggregating 189,818 gross tons, were built during the year ended June 30, 1914, as compared with 121 steel steamships aggregating 225,467 gross tons built in the previous year. In 1914 66 percent of the steel steamship tonnage was built on the Atlantic coast, 7 percent on the Pacific coast, and 27 percent on the Great Lakes, while in 1913 62 percent of the steel steamship tonnage was built on the Atlantic coast, 7 percent on the Pacific coast, 30 percent on the Great Lakes and 1 percent on the Western rivers. The principal vessels built during the last fiscal year are four American-Hawaiian Company steamships, each of 6,600 gross tons, and three Grace Company steamships, each of 6,300 gross tons, all for the Panama Canal trade, and averaging 4,000 net tons. The steamship *Matsonia*, for the Hawaiian trade, 9,728 gross tons, is the largest vessel built in the United States since 1905, and the *John D. Archbold*, 8,374 gross tons, is the largest oil tanker yet built in the United States. On the Great Lakes ten steamers of over 1,000 tons each were built, the *Alton C. Dustin*, 7,978 gross tons, being the largest. Only three large schooners were built. In all, 38 vessels of over 1,000 tons each were built aggregating 169,000 tons.

LLOYD'S SHIPBUILDING RETURNS.—According to Lloyd's returns, which take into account only vessels of 100 tons and upwards, the construction of which has actually begun, there were, excluding warships, 477 vessels of 1,722,124 tons gross under construction in the United Kingdom at the close of the quarter ended June 30. The tonnage now under construction in the United Kingdom is about 169,000 tons less than that which was in hand at the end of last quarter, and over 281,000 tons less than that building in June a year ago.

Questions and Answers for Marine Engineers

Inquiries of General Interest from Readers Regarding Marine Engineering and Shipbuilding Will be Answered in This Department

CONDUCTED BY H. A. EVERETT*

Q.—I would like an expression of opinion regarding the inclosed cards (Fig. 1), showing what defects are indicated and how they should be remedied and also the horsepower of each card. The go-ahead and back-up eccentrics on this engine are cast in one piece, and are keyed on the shaft, so it would be impossible to move them to correct any error. The cut-off is a notched quadrant on the reverse engine, which gives each cylinder the same cut-off. The cylinder diameters are 24, 38 and 61 inches; the stroke, $3\frac{1}{2}$ feet. The piston rods are 5 inches in diameter. The high-pressure cylinder is located forward and has an indirect piston valve. This exhausts directly into the intermediate-pressure steam chest; the intermediate-pressure valve is a slide valve; there is also a slide valve on the low-pressure cylinder. All have direct valve motion of the Stephenson link type. The eccentrics are keyed on couplings which are located between each engine. The cranks are set at 120 degrees.

M. M. P.

A.—The tabulation for indicated horsepower follows:

Cyl.	End.	A. In.	Lth.	Spring	M.E.P.	A. Piston.	I. H. P.	I. H. P. Per Cyl.
H. P.	Top	3 24	4 80	80	54 0	415 5	183	
H. P.	Bot.	3 14	4 72	80	53 2	393 9*	172	355
I. P.	Top	1 08	3 92	40	20 2	1075 2	177	
I. P.	Bot.	1 00	3 89	40	20 5	1015 6*	177	354
L. P.	Top	3 58	4 87	10	7 35	3019	181	
L. P.	Bot.	3 57	4 82	10	7 4	3000 *	181	362

*Assuming P. Rod = 1" Dia.

The distribution of power is excellent and the cards show no serious defects except for the high-pressure cylinder. In this

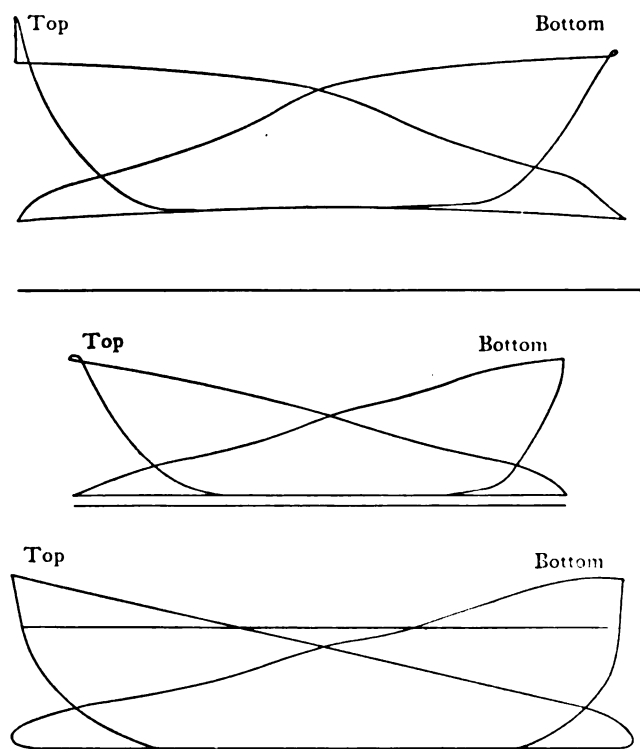


Fig. 1.—Indicator Cards Shown at Reduced Scale

the compression occurs too early on both top and bottom cards, indicating too long exhaust laps of the high-pressure valve and slightly too much angular advance. The exhaust lap for the top end should be shortened and the angular advance slightly reduced. This latter could be done by making an offset key for the eccentric. The intermediate cards could be improved a bit by lengthening the valve stem, which would

* Assistant Professor of Naval Architecture and Marine Engineering, Massachusetts Institute of Technology, Boston, Mass.

retard compression for the top card and accelerate compression for the bottom card. (The valve gear may have sagged.) The curve in the back-pressure line of the high-pressure cards indicates a small first receiver, a restricted flow through the exhaust ports. This latter would be helped by the change already suggested for the high-pressure valve. The whole engine is linked up rather more than customary, but the distribution of power is so good it is doubtful if opening it out would improve matters much.

Q.—Please explain the difference between real and apparent slip and show how each may be calculated.

F. P. A.

A.—The face of a propeller blade is in reality a portion of a screw thread of a certain pitch. Imagine that this thread could be given a certain number of turns in a fixed steel nut having the same thread. It is evident that it would advance a distance equal to the pitch times the number of turns. When working in water, however, the same number of turns would *not* cause the screw thread to advance the same distance, for the reason that the water does not act like the fixed nut. The pressure of the screw upon the water forces some of the water out of the way, thus allowing the screw to slip back a certain amount for each turn. For a given time, then, a propeller working without any slip would travel a distance equal to its pitch times the number of revolutions it has made. As a matter of fact, since the propeller is attached to a ship, it has apparently traveled the same distance that the ship has in that time. The difference between these two distances in a given time is called the *apparent slip*. A propeller having a pitch of 15 feet and making 100 revolutions per minute would advance 1,500 feet per minute without slip. If the speed of the ship is 12 knots, she would travel $12 \times 6080/60 = 1,215$ feet per minute. The apparent slip is therefore $1,500 - 1,215 = 285$ feet per minute, or $285/1500 = 0.19$, or 19 percent.

The real or true slip of the propeller is somewhat different from the apparent slip, in that the passage of the ship through the water causes some of the water astern of the ship to follow the ship. This current of water is called the *wake*, and its speed is dependent upon the shape of the afterbody of the ship. Since this current is moving in the same direction as the ship, the distance that the propeller advances through this moving water in a given time will be less than if the water were still. The slip through this moving water, or wake, is called the *real or true slip*, and is obviously greater than the apparent slip. Suppose that the wake had a speed of 10 percent of that of the ship,* or 122 feet per minute. The propeller would then advance through the wake at a speed of $1,215 - 122 = 1,093$ feet per minute, and the real slip would be $1,500 - 1,093 = 407$ feet per minute, or $407/1500 = .27$, or 27 percent.

The relation between real and apparent slip may be expressed as follows:

$$1 - S_r = (1 - S_a) (1 - W)$$

Where S_r = real slip,

S_a = apparent slip,

W = wake. For the case above noted,

$$1 - S_r = (1 - .19) (1 - .10) = .81 \times .9 = .729$$

Therefore $S_r = .27$.

* The wake can be quantitatively determined for a given slip only by model-tank experiments. It generally ranges from 10 percent to 20 percent.

Q.—What is the best type of fuel oil burner for marine work?
B. O.

A.—There are three types of burners for oil fuel, the classification being made with regard to the method used in atomizing the oil. These three methods are atomization by steam, air or mechanical means.

The steam atomizing burners use 4 or 5 percent of the steam generated by the boilers and, therefore, are not suited for marine use on account of the fresh water required to make this amount of steam.

Air atomization requires large, heavy air compressors to give air under sufficient pressure for atomization. This method requires considerable steam to run the air compressors and also the compressors take up much room and need special attendance.

Mechanical atomization is accomplished by having the oil in the burner under high temperature and pressure. The shape of the nozzle and the internal fittings of the burner are important factors in the atomization. Oil pumps are used to give the required pressure (200 pounds per square inch) and these can be attended by the regular fireroom force.

Mechanical atomization is, therefore, the only practicable method for use with marine boilers at present, and is the only method used at sea by ships of the United States Navy. Then the draft is on the closed stoke-hold principle and the amount of air supplied to the burners is regulated by means of air-cones. In some cases the slots in these cones are made adjustable, in others they are fixed, and the air supply regulated by changing the speed of the fireroom blowers. In the latter type the strips forming the slots are twisted to form guide vanes so as to give a rotary motion to the air. The ends of the burners are fitted with radial slots or tapered screw threads to give a whirling motion to the oil spray, which then enters the furnace in the shape of a cone.

There are five types of mechanical atomization burners at present in use in the United States Navy, viz., Normand, Thornycroft, Schutte-Koerting, Peabody and Fore River.

Q.—Will you kindly trace for me the losses in the power line of a steam-engine-driven ship from the indicated horsepower to the power which is actually used for propulsion, say the power required to tow her from a tow-line?
W.

A.—Indicated horsepower is the power developed in the cylinders of the engines, and as only a portion of this power is finally used in driving the ship, it is important to know wherein the losses occur.

First, the engine itself absorbs a certain amount of power on account of friction of the moving parts. This friction is classed under two heads; viz., "initial friction," due to tightness of pistons, stuffing-boxes, bearings, etc., and "load friction," due to the load upon the bearings, thrust block, etc. The power required to drive the pumps attached to the main engines is classed with initial friction. The amount of power required to overcome these losses varies considerably. The initial friction in large well-adjusted engines with independent pumps is about 3 or 4 percent of the indicated horsepower, whereas 10 percent is required for poorly-adjusted engines having pumps attached. Not much is known about the power absorbed by load friction, but it is generally estimated to be about 7 percent of the remainder obtained by subtracting the initial friction from the indicated horsepower. For a reasonable approximation we can say that friction causes a loss of 10 percent in first-class reciprocating engines having independent pumps, whereas a loss of 15 percent is to be expected in engines having pumps attached. Thus, for reciprocating engines the horsepower actually delivered to the propeller is about 85 to 90 percent of the original indicated horsepower. For turbine engines the horsepower delivered to the propeller is measured directly by a torsion meter placed well astern of the thrust block, the small amount of friction in the bearings aft of this being neglected.

Of the power that reaches the propeller a portion is wasted by the losses of the propeller. The remainder is the power delivered by the propeller to the thrust block. [There is a small amount of power obtained from the wake which is also delivered to the thrust block.] Of the power delivered to the thrust block, a portion is used to overcome the increase in the resistance of the ship due to the suction of the propeller on the hull, and the remainder is the "effective horsepower" or the net horsepower used to drive the ship; this latter corresponds to the power determined by model experiments. The summation of these losses shows that about 40 to 50 percent of the indicated horsepower is lost, so that the effective horsepower is about 50 to 60 percent of the indicated horsepower.

This relation
$$\frac{\text{indicated horsepower}}{\text{indicated horsepower}}$$
 is called the efficiency of propulsion.

Q.—What is the best way to install the internal feed pipes inside a Scotch boiler? Should the feed enter the top or bottom of the boiler? What direction should the water be discharged from the feed pipe to get the best results?
H. F.

A.—In general, the feed valves cannot be placed on that part of the boiler best suited to receive the feed water, so that internal feed pipes are necessary to lead the water to a place where there is a down current of water, in order that the comparatively cool feed water does not interfere with the natural circulation. It is customary to fit two feed valves, one for the main feed pump and one for the auxiliary, the valve for the latter ordinarily being on the left side of the boiler. These valves are of the stop-check type and are placed about on a level with the furnaces. The internal feed pipes should be of brass, and lead upward from the valves between the corrugations of the furnace and along the shell of the boiler to a point somewhat above the level of the tubes, but still below the lowest working level of the water. From here they are led back into the boiler about two-thirds of the length, and then should be turned downward somewhat in front of the combustion chamber. This length of pipe also provides a means for further heating the feed water. Care should be taken that steam cannot enter the internal pipes, as the sudden condensation of the steam by the cool feed water produces shocks which may damage the piping. The internal pipes should not touch the shell except where they connect with the external fittings.

It is not a good plan to discharge the feed water at the bottom of the boiler, as the water spaces under the furnaces do not receive sufficient heat to provide good circulation there, and the entrance of cool feed water would retard this rather than assist it.

In some cases the feed water has been introduced into the steam space in the form of spray, but while this plan has some advantages, it has not met with general favor. It cannot be used in boilers liable to prime, and also may cause the steam pressure to fall when the feed is turned on or increased.

NEW BOMBAY DREDGER.—The Bombay Port Trust, through their consulting engineers and agents, Messrs. Sir J. Wolfe, Barry & Partners Company, recently placed an order with Messrs. William Simons & Company, Ltd., Renfrew, for a powerful bucket hopper dredger, similar to the *U. S. Corozal*, built by the same company for the Panama Canal. The new dredge will be used for deepening the Suez Canal at the Port of Bombay and the approaches thereto.

UNVEILING OF MEMORIAL TO TITANIC ENGINEERS.—The ceremony of unveiling at Southampton the memorial to the engineers of the *Titanic* was performed April 23 by Sir Archibald Denny before a large assemblage. The address was delivered by Albert W. Swalm, United States Consul at Southampton.

Letters from Practical Marine Engineers

Incidents Relating to the Design, Care and Handling of Marine Engines, Boilers and Auxiliaries; Breakdowns at Sea and Repairs

An Air Pump Breakdown

Several years ago, when the writer was serving as assistant engineer on a large coastwise vessel, the main engine, which was a condensing triple with independent air pump, suddenly slowed down. An investigation showed that nothing was wrong with the engine or the boilers, so that the natural conclusion was that something was wrong with the exhaust, as the vacuum was suddenly lost in the condenser.

Upon investigation it was found that the trouble was in the air pump, so the engine was stopped and the pump quickly taken down. It was found that the pump rod had broken at

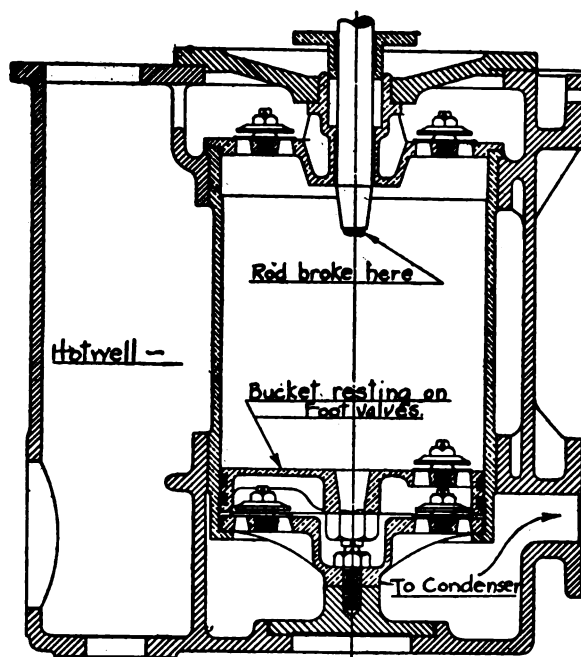


Fig. 1

the end of the threaded part, causing the bucket to drop off and rest on the foot valves, as shown in Fig. 1. The bucket and rod were then removed. The nut, which had dropped off with the broken piece of the rod still in it, was also taken out and the cover plate replaced. The gland in the cover plate was then plugged with a soft wood plug, which swelled up and filled the gland nicely, so that it did not leak. The circulating pump was a force pump and could not be used to suck through the hotwell. It was, therefore, cut out and the auxiliary pumps thrown in so that they drew direct from the hotwell. This only meant the turning on and off of valves, so therefore did not require much time, and the engine was soon started up and worked fairly well with such a temporary arrangement.

However, the trip was a long one and the rod had to be repaired, otherwise there would be a decided loss of speed and economy. Fortunately the vessel was equipped with a fairly good machine shop, so that the repair could be done carefully.

The end of the rod was faced off down to the bottom of the taper. It was then drilled and tapped to take a $1\frac{3}{4}$ -inch standard bolt. A counterbore was then made about $\frac{1}{2}$ inch deep, with a diameter equal to the diameter of the bolt ($1\frac{3}{4}$ inches). A steel stud was then made to fit this, as shown in Fig. 2. This was made with a $\frac{3}{4}$ -inch shank having the same

diameter as the counterbore, so that the one would fit the other very tightly. The end that fitted into the rod was made fairly long for strength and rigidity, while the other end was made the same as the original and threaded to take the nut that had been on the rod.

By making the shank fit tightly into the counterbore and making it $\frac{1}{4}$ inch longer than the counterbore a material amount of strength was gained, not only in gripping, but in general rigidity. The broken piece was easily removed from the nut and the bucket and rod replaced while the engine was again stopped.

With the bucket and rod again in working order the engine

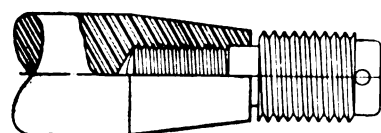


Fig. 2

was started and continued to run as well as it had previous to the breakdown. Naturally, upon reaching port, a new plunger rod was procured, but it was unnecessary to install it at the time. The repaired rod was not taken out and replaced by a new one until the engine was thoroughly overhauled several years later. The piece was found to be in excellent condition and could easily have continued in service for some time.

STUD.

Let Well Enough Alone

From what little experience I have had in the engineering line, it seems to me that a large amount of trouble is caused by continually accepting substitutes for standard goods of known and tried values. This is probably so at the present time as much as at any other, and it is believed that the ability of selling agents will keep it so for years to come. It is only through the general appreciation of the value of a standard article, by both engineers and owners, that much of this trouble can be eliminated.

There are two sides to every question, and in such a question as the adoption of a new brand of material there are two distinct points to be considered in determining whether or not a change in the article in use is justified. One point is concerned with the material, the other with the personal side of the matter.

In regard to the material side, it is only logical to suppose, unless prices are known to be exorbitant, that one is getting good value from the manufacturers, and in such a case it is not very logical to suppose that one is going to get much better values from some other manufacturer. This does not always apply. Perhaps an attempt is being made to introduce a new article of merit. Such an article may be sold at a very low price in order to introduce it, but when once established it must be soon sold at a price that will offer a fair profit to the makers, so such a gain will usually be only temporary. In such a case one may have made a small monetary gain if the material has been satisfactory. Considering that the material may not be satisfactory, and may be almost a total loss, the best that may be said is that such a change, from a monetary point of view, is little more than a gamble.

It is also well to remember, in regard to material, that the average job on board ship is small, and does not represent a very large investment in material. Take, for instance, the

renewal of packing in a valve stem or in a steam pipe flanged joint. A difference of value of 5 or 10 cents ($0/2\frac{1}{2}$ or $0/5$) in the packing used for the job seems quite immaterial when the value of the labor and the possibility of delays due to blow-outs, etc., are considered; yet it is just such cases as this that usually cause trouble, for when that additional 5 cents ($0/2\frac{1}{2}$) is multiplied by the total number of gaskets that can be cut from the whole sheet of packing, it makes the extra cost of the whole sheet seem much too large. The result is that sometimes an inferior brand is purchased and that a blow-out causes more expense from damage and delay than a year's supply of the high-grade packing would have cost.

I am quite convinced that the personal side of the question is just as important as the material side. The average man aboard ship cannot be expected to take kindly to the idea of discarding what he knows to be good from experience in order to adopt something that he knows nothing about. The man aboard ship is desirous of making a job that will hold; he does not want to be obliged to renew it again in a few days.

It is this natural tendency that makes a known good material go much farther than an unknown material of equal merit. Among many cases I recall one in which the engineer's force had been using a packing called T— with the very best of results. This packing was expensive, and it was therefore decided to substitute another that was much cheaper and that gave equally good results on test. The results were not what might have been anticipated. The cheaper packing cost more in the end than the T— originally used, and this purely because of the personal side of the question. The new brand was not used with such economy; since the men did not have faith in it it was always being renewed, where the old and tried packing would not be renewed, and in the end the greatly increased consumption alone made the total cost of packing smaller for the high-grade packing.

In the use of oil, I recall one case in which a sea trip had been habitually made on 40 gallons of oil. Each oiler knew just what this particular brand of oil would do and placed great faith in it. This oil was made by a trust, and was believed to be entirely too expensive, so another oil, said to be equally as good, was taken on board. This last oil was not made by a trust, and was not so expensive for that reason, according to theory. The result was that it required 70 gallons of the new oil to do the same work that was done by 40 gallons of the old oil. The next result was a return to the trust-made oil and also to the consumption of 40 gallons per trip. Of course, more experience with the cheaper oil would have materially lessened the amount needed, but it is questionable whether its total cost would ever have been as low as that of the higher-priced product then in use. Substitution would have therefore been a very poor policy on account of the increased bulk if for nothing else.

The number of these cases could be increased almost indefinitely, but these two fully illustrate the points mentioned: in the first the personnel was probably responsible for the lack of economy; in the second the fault was in the material itself. In neither case was there anything gained by the change.

I do not think of advocating the use of high-priced material when equally good results may be obtained for less money, nor do I intend to discourage experimentation with new materials, but I do desire to impress upon the engineer that the old motto, "Let well enough alone," has its application to marine engineering as well as elsewhere, and that changes should be made with care and with due consideration of everything involved.

The claims of agents and of advertising matter are, in many cases, so much greater than the material can possibly fulfill that they should be regarded with great suspicion. A standard line of goods, thoroughly tried out, giving good satisfaction, and enjoying the confidence of the personnel of the engineer's department, is not to be lightly discarded even for

the saving of a few dollars per year. It will sometimes pay to refer the claims of the agent selling new goods to the manufacturers of the standard line now in use for their comments. The manufacturers will be pleased to comment, and may perhaps give some very good advice in the matter.

W. W. B.

A New Method for Obtaining "Offsets"

The writer was called upon to make the necessary preparations for an inclining experiment on a certain vessel the interior arrangements of which had been materially changed at various times after the vessel was launched. Information regarding the vessel's displacement at varying drafts was not available, nor were any of the necessary curves which are used in these and similar ship calculations. Even the body plan was missing, therefore offsets for the body plan had to be obtained, and this work had to be done as quickly as possible, because the ship was then in dry dock and such work can only be done while the ship is in dry dock.

The usual method of obtaining these offsets is shown at the left in Fig. 1. A plumb line is first suspended at the centerline of the ship to see if the ship is absolutely vertical in the

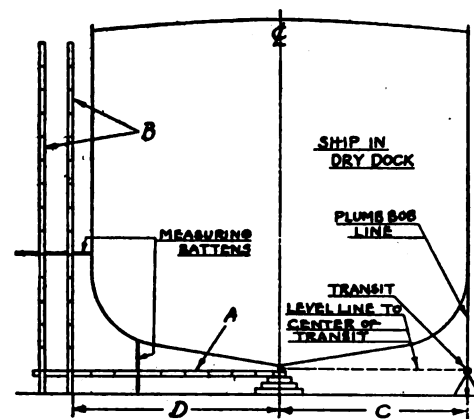


Fig. 1

dock. If not, the amount of deviation from the vertical in a certain distance is noted, and this is used in setting up the batten *A* at right angles to the vertical centerline of the vessel.

For simplicity we shall assume that the ship is vertical, then batten *A* is fixed in a perfectly horizontal position at the bottom of the keel. This batten is marked off about every two feet and the divisions numbered. The zero mark should coincide with the centerline, i. e., the center of the keel. Just beyond the extreme half-breadth two vertical battens *B* are erected. These battens are divided in the same manner as batten *A*, and their zero marks should coincide with the edge of *A*. It is evident then that with the aid of measuring battens the offsets can be obtained, referred to the vertical and horizontal battens as shown, for various sections of the vessel, and from these a body plan can be drawn and faired in.

This method requires all battens to be carefully divided, the battens *A* and *B* by marks 2 feet apart and the measuring battens by eighth-inch intervals, as on a common scale measuring inches and eighths.

For the case in question, however, such battens and means for holding them in their respective positions, etc., were not directly available, so that the following method was adopted to obtain the offsets:

Various stations were located for convenience at the edges of deck houses, etc., and by squaring across to the rail two plumb lines were dropped from the same point. One of these was just long enough to reach the flooring of the dock from any point of the rail, while the other was long enough to

girth the largest section of the vessel from the rail to the keel. The string that was to girth the section was then drawn in and fastened to the turn of the garboard strake, or upper edge of the keel, by drawing it down behind a heavy iron bar, the upper end of which held the string in close contact with the hull.

From the plumb line any convenient base line was laid off parallel to the ship's keel on the flooring of the dock. The base line was usually from 30 to 40 feet long, depending on obstructions. At the other end of the base line an engineer's transit was set up, so that its horizontal axis would come in the same horizontal plane with some point on the keel. This point was found for each section, so that all the sections could be referred to a common base line.

To find the point, the "gun" was swung around and used as a level, while an assistant marked the keel in the line of sight.

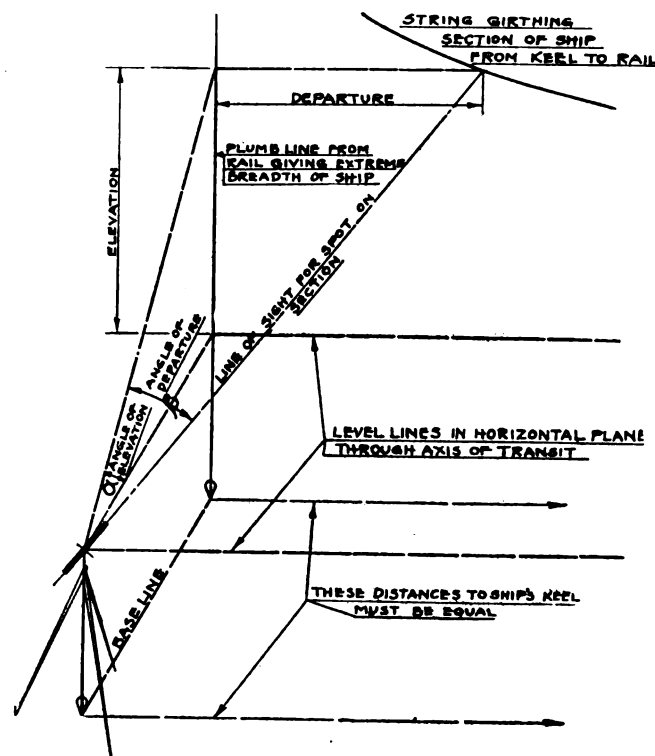


Fig. 2

The "gun" was then swung back and lined up with the plumb line, and held while the horizontal scale was turned until its zero reading coincided with the zero of the vernier. This made it easier to handle the readings in the office, as they all read from zero at each section.

It only remained to take an elevation and a departure, each in degrees, for enough spots on each section to be able to lay the spots off on the drawing and run a fair section through them. We were always careful to take the edge of the keel where the string crossed it as one spot, so that we could determine the half breadth of each section, knowing the thickness of the keel.

A fairly good idea of what was done is represented in Fig. 2. This is a sketch showing the necessary lines and distances in perspective, i. e., as it would look to a person on the scaffolding about one-third the way up the ship's side.

The readings were kept systematically, so as to facilitate the work in the office, where the triangles were solved to get the actual elevation and departure in feet, inches and eighths of an inch instead of degrees, minutes and seconds. The elevation is equal to the base line length multiplied by the tangent of α , Fig. 2. The departure is equal to the base line multiplied by tangent $\beta / \cosine \alpha$. When these are calculated and tabulated for each section it becomes a simple matter to

change them all to a common base line and lay out each section.

The sections can be located longitudinally from the deck plan, as previously stated. In this way the full sheer draft may be drawn and faired by waterlines, buttocks and diagonals. Then equidistant sections may be laid out and a new body plan drawn showing these. This should be done at a large enough scale to admit of accurate calculations.

In using such offsets it will be seen that they are taken to the outside of the outside strakes throughout. Allowance can be made for this so that the sections will all be drawn half the thickness of the shell plating inside of this, giving a displacement, etc., from the body plan so constructed that will be exact.

Of course, in using this system difficulties will be encountered if some sections have double curvature. In that case either avoid them and trust to fairing or use a little ingenuity in attaching the string. "TRANSIT."

An Emergency Pump Rod

On one occasion the writer had the experience of repairing a bilge pump on a large yacht. While on a rather extended cruise the bilge pump rod broke beyond repair, and it became necessary to do something, as the pump was needed continually. In looking for a solution to the difficulty it was noticed that the vertical columns on the independent feed pump were of the same diameter as the rod of the bilge pump. The feed pump was therefore dismantled sufficiently to take out one of the columns, which was then turned up and fitted in the bilge pump. Very little machining had to be done on it, since it was of the same diameter as the original rod. It was necessary, however, to replace the column in the feed pump to prevent further and more complicated difficulties.

This was done by using a couple of pieces of live oak of the same length as the distance between the shoulders on the column. These were clamped together with a steel rod between them, which had been cut just the same length over all as the original column. The rod was threaded on the ends and nuts were made for it.

It is readily seen that when this was set up as a column the live oak struts were sufficient for compressive stresses, while the rod held them in place and took care of all the tensile stresses brought to bear on the cylinders.

This arrangement not only proved to be a good temporary repair, but both the oak struts and the rod have been in service for over a year. Of course, new parts have been procured to be ready for any emergency, but the incident shows what good, substantial repair work can sometimes be done in case of emergencies, especially at sea. Incidents like this are not only interesting but instructive when related, and should be a great help to those who may at some time be up against the same or a similar proposition.

LIVE OAK.

THE BOLINDERS COMPANY-NEW YORK.—The Bolinders Company-New York, with offices at 30 Church street, New York City, has been formed to manufacture and sell in the United States the two-cycle, valveless, direct-reversible, crude oil engines of the type developed by J. & C. G. Bolinders Mek. Verkstads A. B., Stockholm, Sweden. These engines are built in sizes ranging from 5 to 320 horsepower, and up to the present time over 10,000 of them have been placed in service.

THE NAVAL APPROPRIATION BILL.—The total naval appropriation bill, as passed by the Sixty-third Congress for the coming year, amounts to \$140,350,833.61 (£28,800,000). The programme for new construction for the coming year includes two battleships, six destroyers and eight or more submarines, while the recent sale of battleships *Idaho* and *Mississippi* to the navy of Greece makes available an additional fund for the construction of a third first-class battleship.

Marine Articles in the Engineering Press

Problems in Naval Architecture.—In a series of three lectures on modern ships, just concluded at the Royal Institution by Sir John Biles, the main problems in naval architecture were grouped under three general headings: Smooth water sailing, ocean travel and the war navy. Under the first head the problems of buoyancy, stability, strength and resistance of ships were discussed from the viewpoint of the naval architect. The discussion of propulsion, which included both steam and sailing ships, covered the relations of effective propulsion to machinery design and the methods of investigating resistance, power and propeller efficiency. The second lecture, devoted to ocean travel, discussed the problems of speed, comfort and safety, subdivision and the attendant calculations for stability. The third lecture, dealing with the war navy, showed clearly that the same problems of stability, strength and subdivision apply to warships as well as to other ships, except that provision has to be made for a greater extent of flooding in the warship. 5,250 words.—*Engineering*, March.

Development of Internal-Combustion Engines for Marine Purposes.—By M. Peloux. This paper describes the improvements in the type of Diesel engines manufactured by Messrs. Schneider & Company for marine purposes. In general, the improvements enumerated relate to the starting and maneuvering gear, the exhaust, air-inlet, starting and fuel valves and the fuel pumps, air compressors and means for lubrication. The starting and maneuvering gear is arranged with a view to starting and reversing the engine as easily and as quickly as possible. One complete revolution of the driving shaft suffices to start the engine. This is done by hand, through a reducing gear or by a Servo motor. Valves in the cylinder heads are provided with special devices to prevent their falling into the cylinders if their rods break. The fuel valves are either direct-acting straight needles or are balanced and work by rotation. The fuel valve casing of the cylinder is fitted at the bottom with a ball pulverizer and a multiple hole diffuser. Each oil pump is regulated by varying the amount of lift of its suction valve. 13 illustrations.—*The Institution of Mechanical Engineers*, June.

The 32,000-Ton Floating Dry Dock at the Portsmouth Dockyard.—The new floating dry dock at the Portsmouth dockyard, designed to take the largest and heaviest naval vessels in service, was built by Messrs. Cammell, Laird & Company, Ltd., Birkenhead, to the designs of Messrs. Clark & Standfield. The dock is of the "box" type, and consists of a pontoon or lifting portion and two parallel side walls built on to and forming part of the pontoon. When floating at its moorings in sea water the dock is capable of lifting and carrying a vessel of 32,000 tons displacement with drafts up to 36 feet, the weight of the shores and berthing gear being taken as part of the weight of the ship. The dock is 680 feet long over all and 144 feet wide. The clear width between the rubbing timbers at the top deck is 107 feet 6 inches. The side walls are 65½ feet high on the outside and rise 40½ feet above the pontoon. The depth of the pontoon is about 20 feet. Each of the side walls has a watertight deck running the whole length, and the bottom pontoon is divided both longitudinally and transversely by a number of watertight bulkheads. These bulkheads and decks divide the pontoon and walls into about 80 watertight compartments. The displacement of the dock up to within 6 inches of the pontoon deck at the center is 47,400 tons. The total weight of the dock, including 3 inches of water in the pontoon, is 15,400 tons, and thus the net lifting capacity is 32,000 tons, giving 207 tons per inch immersion in way of the pontoon. The mini-

mum metacentric height of 5 feet 6 inches occurs when the water is about 4 feet above the top pontoon. This increases to 47 feet 6 inches when the pontoon deck rises from the water. The pumping installation consists of eight Babcock & Wilcox watertube boilers, together with eight sets of centrifugal pumps of the vertical spindle type, each set being driven by a compound condensing engine. The boilers are placed two at each corner of the dock with the pumping engine rooms separated from the boiler rooms by watertight bulkheads. The electrical equipment consists of four direct-coupled generators, each having an output of 900 amperes and 225 volts. A Westinghouse motor generator is fitted in each wall, having an output of 1,000 amperes, for the purpose of supplying current to a ship in the dock. Workshops are arranged in the starboard wall of the dock. 12 illustrations. 2,000 words.—*Engineering*, May 1.

The Motor Tank Ship Elbruz.—A description of a motor-driven oil tanker, built to the order of the Société Anonyme d'Armement d'Industrie et de Commerce, of Antwerp, by the Tyne Iron Shipbuilding Company, Ltd., of Willington Quay-on-Tyne, the propelling machinery being supplied by the Nederlandsche Fabriek van Wertuigen en Spoorwegmaterieel, of Amsterdam. The vessel is 300 feet long over all, 51 feet 2½ inches extreme beam, 29 feet molded depth, and at a draft of 23 feet displaces 9,400 tons. The hull is constructed with transverse framing, with the side stringers continuous in way of the transverse bulkheads. The machinery, which is placed aft, consists of two reversible single-acting six-cylinder Werkspoor engines, with cylinders 22 inches diameter and 30¾ inches stroke, working on the four-stroke cycle and driving twin screws. The engines develop a total of 2,200 brake horsepower when running at 125 revolutions per minute. The auxiliary machinery is steam driven, two cylindrical multitubular donkey boilers being installed, one in the engine room and the other on the deck above. Both boilers are arranged for burning liquid fuel on the Meyer-Smith system, while the lower boiler is also arranged for heating by the exhaust gases. 9 illustrations. 1 plate. 900 words.—*The Shipbuilder*, June.

High-Degree Feed-Water Preheating on Steamers.—By Offerdinger. The author presents a new method of preheating boiler feed-water on steamers by using the exhaust steam from auxiliary engines and receiver steam to a temperature of 284 degrees F. and higher. The advantage of such high preheating of feed-water lies, first, in the improvement of the efficiency of the engine plant, and, second, in the beneficial action on the life of the boiler, which is in this way spared from the heavy stresses consequent upon the injection of comparatively cold water. In addition, the higher the temperature of the water the freer it is from air, which is objectionable when present in the boiler. For such high preheating two-stage preheaters in series are used, working on steam of different pressures. When used in connection with triple or quadruple expansion engines the feed-water may first be heated by the exhaust steam from the auxiliary engines with an addition of steam from the low-pressure receiver. From this preheater the water passes into the second element, where it is heated by the steam from the intermediate pressure receiver, the pressure of the steam in the second element being maintained at a higher level than in the first preheater. Data are given from tests showing the results of feed-water preheating in one and two stages. The saving in heat by using the two-stage preheater arrangement as compared with the single-stage arrangement amounted to 3.5 percent. 2 illustrations.—*Zeitschrift des Vereines Deutscher Ingenieure*, April 18.

Recent Japanese Warships.—By Rear Admiral Constructor Motoki Kondo. A paper read before the Japanese Institution of Naval Architects at Kobe. The remarkable development of the Japanese navy during the last ten years is shown by the completion of six battleships, five battle cruisers, five second-class cruisers, seven gun boats and thirty-nine destroyers, not to mention torpedo boats, submarines and other vessels of smaller size. The main particulars of the types of vessels added to the Japanese navy during the last ten years are enumerated and it is pointed out that, together with the development of the navy, the facilities for naval construction in Japan have likewise been steadily improved. The mild steel used in Japanese warships is now almost entirely supplied by the steel works at Kiushiu and, in addition, a number of ship fittings, auxiliary machinery, etc., are now obtainable from Japanese manufacturers. 22 illustrations. 3,000 words.—*Engineering*, March 27.

The Motor Ship Arum.—This article includes a detailed description of the engines and machinery arrangement of the motor ship *Arum*, a twin-screw general-cargo ship built by Swan, Hunter and Wigham Richardson, Limited, for Sir Marcus Samuel, for the Persian Gulf trade. She is 360 feet overall, 47 feet beam and 27 feet molded depth, and will carry about 5,000 tons deadweight at about 10½ knots. The engines are of the four-cylinder Polar Diesel type built by the same firm. Each set of engines has four cylinders of 16.15 inches diameter by 33.9 inches stroke, and gives 675 brake horsepower at 123 revolutions per minute. The fuel valve rocker-arm is the only moving part visible in the valve mechanism, except, of course, the camshaft. A light cast iron casing is fitted over the shaft, the only opening being a narrow slot for the rocker to work in. The lower part of this forms a trough in which an oil bath submerges the rollers and leads to a very quiet valve gear. The method of driving the valve gear is rather unique and is composed of a pair of "coupling rods," from eccentrics carried on a disk on the forward coupling of the crankshaft. Another feature of the design, and an important one, is the method of supporting cylinders, leaving the lower part of the engine considerably open, since there is no forced lubrication. The exhaust is led into big water-jacketed pipes, and at the end of the engine room can either be passed up the funnel or under the donkey boiler. All the auxiliaries are steam-driven and they include an auxiliary air compressor, electric lighting engine, turning engine, and pumps. 6 illustrations. 2,200 words.—*The Engineer*, May 29.

Development of High Power Marine Diesel Engines.—By James Richardson, B. Sc., Assoc. M. Inst. C. E. The progress of the marine Diesel engine, although remarkable thus far, is temporarily at a standstill, awaiting the application of the experience gained during the first phase of its evolution. There are several causes for such a condition and probably the most important are: Over-ambition in increasing the sizes of units, using land engines in the marine field without making necessary modifications, and the lack of experimental and research work. The advantages of the Diesel engine in the marine field are well known, but there is some debate as to the relative merits of the two general types, the two- and four-stroke cycle. In considering future possibilities Mr. Richardson treats only of the two-stroke cycle, and throughout the discussion he deals systematically with each part, its functions and possible improvements. In Diesel engines the speed of revolution is relatively higher than that of the steam engine, and this is allowable because of the necessary accuracy of workmanship and the excellence of material required by the parts subject to pressure and temperature. The piston speed brings up the question of inertia, which, if high enough, permits of easier lubrication and varies the direction of pressure on the main bearings. The stroke-bore ratio depends on the ratio of combustion volume to cooling

surface, method of scavenging limits in height of engine and the type of crankshaft adopted. Mean effective pressures average about 65 pounds per square inch on a brake horsepower basis for normal running. Various means by which this mean effective pressure in the future may be augmented are given and methods of accomplishing same proposed. An investigation of the loads on engine parts, particularly on crosshead guides and columns, of an oil engine and a steam engine, both of the same power, etc., shows some marked differences and forms a very good basis for design in comparing with steam engine practice. Very many types of engine framing are used and all prove more or less efficient. Crankshafts and crank-webs involve thorough investigation, and at best so many variables are brought into formulas that a certain amount of experience is necessary in designing these parts. The question of cylinders and cylinder-heads is probably the most varied one in the entire design. Assuming that the material used throughout is excellent, as it should be, the arrangement and design of the cylinder and cylinder-heads, with their various valves and ports, determine to a large degree the efficiency of the engine as a whole in its cycle of operations, power developed and fuel economy. Cast iron is almost universally adopted for the cylinder-heads and liners, cast steel having proved unsuccessful for such purposes. In two-cycle engines, whether Diesel or of the ordinary internal combustion type, scavenging is an advantage and means a gain of power. In the Diesel two-cycle type scavenging is necessary and a scavenging air pump is fitted to supply air at from four to eight pounds per square inch to clean the cylinder of the products of combustion. For large engines of the future the methods of using two valves or two ports will prove the best. Two and three stage compressors are used for purposes of starting, reversing, etc., and supply air at pressures from 300 to 1,000 pounds per square inch. As to fuel injection, since the combustion in two-cycle engines takes place so rapidly, the timing of injection must be very accurate and regulation for slow running is difficult. The design of valve gears should not present difficulties to the engineer, because something definite is required and variation of temperature and pressure does not materially affect these parts. Starting by compressed air is the practice almost universally adopted, and for this purpose it must be kept on hand or arranged for by auxiliaries. 49 illustrations. 10,000 words.—*Engineering*, April 24 and May 1.

The Berlin-Stettin Canal.—A new German waterway, officially termed the "waterway for large ships, Berlin-Stettin," an extension of the one already existing, has just been opened. This canal, one of the many large and costly canal constructions lately carried out by Germany, forms a new waterway between the capital and the most important sea-port of Prussia. Much difficulty was experienced not only in various stages of construction, but at the very outset arguments were advanced as to the better of two routes. While all this discussion was going on it became urgent to start the construction, as the existing Finow Canal proved less and less capable of handling the rapidly increasing traffic. The new canal or waterway has a length of about sixty-two miles. Loading places and private wharves have also been constructed at the expense of municipalities and other bodies. Quays have also been built, especially where Crown lands adjoin and where the sides of the canal have been puddled. The towing on the canal will be done by means of steam tugs, and it is to be compulsory for all vessels not self-propelled which traverse the canal. 5 illustrations. 3,000 words.—*Engineering*, May 8.

BOILER MANUFACTURERS' CONVENTION.—The American Boiler Manufacturers' Association will hold its twenty-sixth annual convention at the Waldorf-Astoria Hotel, New York City, Sept. 1 to 4.

New Books for the Marine Engineer's Library

STANDARD METRIC EQUIVALENT TABLES. Size, 12½ by 10 inches. London, W. C., 1913: The Central Translations Institute. Price, 1/2.

These tables comprise weights and measures, also prices in francs and marks, the latter being based on figures from the Board of Trade for the rates of exchange. The tables, which are neatly printed on a large card, should prove very useful to those who need accurate and authoritative values for conversion.

TIDE AND SPEED TABLES FOR 1914. Size, 2¾ by 4 inches. Pages, 248. London, 1914: Alfred Graham & Company.

Published in handy pocketbook form, this volume gives not only tide tables for the principal seaports of the United Kingdom, port charges, knot tables, distance tables, electrical terms, etc., but also includes some handy engineers' tables, giving the dimensions of chain cables, sizes of tanks and the weights of wrought iron plates, angles and round bars, and also a table for the calculation of the consumption of coal per day based upon indicated horsepower per hour.

PERCENTAGE COMPASS FOR NAVIGATORS, SURVEYORS AND TRAVELLERS. By John C. Ferguson, M. Inst. C. E. Size, 36 by 23 inches. London, E. C., 1913: Longmans, Green & Company. Price, unmounted, 2/6 net; mounted on linen, 3/6 net.

Ferguson's percentage unit of angular measurements, a system of measurement that reduces problems in plane trigonometry to simple arithmetic, has been practically applied to a compass dial to form what is known as a percentage compass. There is no doubt that the practical application of this system to any engineering computations involving angular measurements, etc., will effect a tremendous saving of work and a decided simplification of all calculations in plane trigonometry.

GREAT LAKES RED BOOK. 1914. Size, 3 by 4¾ inches. Pages, 147. Cleveland, Ohio, 1914: Penton Publishing Company. Price, \$1.00.

This book contains a list of over 1,000 vessels on the Great Lakes, with the name of owner, captain and engineer of each vessel. The list is arranged according to the fleets to which the vessels belong, each fleet has a separate number, and preceding the list is an index in which the names of the vessels are arranged alphabetically, with the number of the fleet following the name of the vessel. There is also a list of ore carriers, arranged alphabetically, giving the capacity of the vessels in gross tons on about 19 feet draft, and a vessel supply directory, giving a list of fueling ports, engineers' supply houses, ship chandlers, proctors in admiralty, grocers, butchers and others who furnish supplies or make repairs to vessels at different ports on the Great Lakes.

ANDREW THOMSON'S YACHTING GUIDE AND TIDE TABLES, 1914. By Andrew Thomson, A. M. I. N. A. Size, 3¼ by 4½ inches. Pages, 184. 41 Pall Mall, London, S. W., 1914: Andrew Thomson. Price, 1s.

Beginning with general tide tables giving the times of high water at London Bridge and at various places in England and Wales and along the coasts of Scotland, Ireland, France, Holland, Belgium, Germany, Spain and Portugal, the author goes on to give tables of English equivalents for metric weights and measures and tables of comparison of English and foreign money values. Following this are several pages of general items relating to stowage measurements, signals for pilots, admiralty warrants, registration of yachts and regulations for preventing collisions at sea. Carefully compiled

tables show the distances from English ports to foreign ports, and also time differences between Greenwich and the most important ports of the world. The principal tables in the book, however, relate to rates of speed, time allowances and racing rules of the Yacht Racing Association. There are also lists of cup winners and detailed instructions for the measurement of yachts as drawn up by the Permanent Committee of Berlin, 1906. Although this is a small volume, it contains much valuable information for yachtsmen, most of which is arranged in convenient form for ready reference.

WANNAN'S MARINE ENGINEER'S GUIDE. Fifth edition, in two parts. By A. C. Wannan, C. E., and D. Lindsay. Size, 5½ by 8 inches. Pages, Parts I and II, each 280. Numerous illustrations. London, 1913: Crosby, Lockwood and Son. Price, 6/- net, each.

Part I deals with only the subject of arithmetic and contains a great number of questions from marine engineers, with simple, clear and correct solutions to the same. Notes on beams, boilers and mechanical powers are also included, together with the latest Board of Trade regulations. Part II contains 310 elementary questions with illustrated answers, and verbal questions and answers, together with a set of thirty-seven completed drawings. Indicator diagrams are discussed in detail and each drawing where necessary includes a note giving the particular points and the requisite formulæ for the various parts. Both books are very complete and should prove a most valuable collection of data and information for marine engineers.

THE DOCK AND HARBOR ENGINEER'S REFERENCE BOOK. By Brysson Cunningham, B. E. Size, 4 by 6½ inches. Pages, 320. Illustrations, 144. Plates, 6. Philadelphia, 1914: J. B. Lippincott Company. London, 1914: Charles Griffin & Company, Ltd. Price, \$3.00 net.

REVIEWED BY H. McL. HARDING*

As Mr. Cunningham's book is about the only one of its kind printed in English it is of special interest and value. It is a condensed compilation from such accepted authorities as the Proceedings of the International Navigation Congress, the minutes of the British Institute of Civil Engineers and various technical journals. It also contains many quotations from previous books by the same author.

The method of references is extremely satisfactory, as it enables the engineer to consult readily the original sources of information. As far as is possible the costs of construction of the port works are reduced to a unit basis, such as acreage or lineal foot lengths, with reasons for variations from these unit prices clearly explained. The equipment necessary for carrying out harbor construction and the costs of the apparatus are given in detail.

The general plan of the book is comprised under the following headings: Port Administration, Nautical Data, Methods of Construction and Costs of Harbors, Docks, Wharves, Locks, Graving and Floating Docks, Dredging and Maritime Canals and Coast Protection. There are also descriptions of the engineering work of most of the important seaports, including the often neglected fishery and minor harbor plans.

Excellent judgment has been shown by the author in selecting the data which would be the more valuable for marine engineers, and the book should be on the desk of every harbor engineer, not only as a book of reference, but also as a book of instruction.

* Consulting Engineer, Marine Terminals, New York.

RIVERS AND ESTUARIES OR STREAMS AND TIDES. By W. Henry Hunter, M. Inst. C. E., M. Am. Soc. C. E. Size, 5½ by 9¼ inches. Pages, 68. Illustrations, 8. London and New York, 1913: Longmans, Green & Company. Price, 2/6 net.

The elementary study given in this book is, to a large extent, based on lectures delivered to the students of the Institute of Civil Engineers in various parts of Great Britain, and upon further lectures on streams and tides delivered to the engineering students in the University of Manchester. The object of these lectures was to furnish the young students information, and to offer them counsel, which experience had taught the lecturer would have been of lifelong service to him had he known these facts when he himself was a student. The subjects discussed include river improvement works and estuarial works, giving both European and American examples.

MEXICAN FUEL OIL. Size, 6¼ by 8¾ inches. Pages, 150. Numerous illustrations. London, E. C., 1914: Anglo-Mexican Petroleum Products Company, Ltd. Price, 3/6 net.

While the book is intended primarily to give a comprehensive account of the production, refining, distribution and uses of Mexican oil as produced by the Mexican Eagle Oil Company, it includes a very general discussion on the advantages of oil for fuel and its many and varied uses. Starting with a review of the progress of petroleum, the general advantages of fuel oil and the types of burners and systems used, descriptions are given of special applications of fuel oil in the navy, the mercantile marine, on railroads and for stationary steam plants. All through this discussion prominence is given to the subject of coal versus oil fuel, the data given showing some very startling advantages for fuel oil in practically every one of its many applications.

ARITHMETIC OF THE STEAM BOILER. By Charles J. Mason. Size, 5 by 7 inches. Pages, 225. Illustrations, 21. New York, 1914: McGraw-Hill Book Company, Inc. Price, \$1 net. London, E. C., 1914: Price, 4/2 net.

This book is a compilation of arithmetical rules and formulas applicable to steam boilers of various types. The author claims no originality in the preparation of the material, excepting only the arrangement and manner of presentation. It is intended as a book of reference for those who may require rules and formulas directly related to steam boilers, and its aim is concentration and logical order in the arrangement and treatment of the various features introduced. It is not intended to teach the elements and principles of arithmetic in this book, as might be inferred from its title, but only the application of arithmetic to steam boiler calculations. It is presumed that those who may use it already understand arithmetic but desire to have a compact set of rules and formulas conveniently ready for use, without having to look through several books for a certain formula when required. Those who are preparing for examination for engineer's certificates and licenses will find the work of great assistance to them.

HANDBOOK FOR MACHINE DESIGNERS AND DRAFTSMEN. By Frederick A. Halsey, B. M. E. Size, 8½ by 11 inches. Pages, 494. Numerous illustrations. New York and London, 1913: McGraw-Hill Book Company. Price, \$5.00 net.

After over thirty years of active association with machine design and construction, fifteen of which were spent in designing machinery for one of the leading manufacturers of the United States and eighteen as editor of the *American Machinist*, Mr. Halsey has brought together in convenient form for desk use the essential data and basic facts which designers, draftsmen, superintendents, engineers and machinists need constantly in the work of designing all kinds of machinery and machine parts. The book is in no sense a treatise on machine design or applied mechanics. It is a collection of

material carefully digested, classified and indexed, presented in such form as to give as clearly as possible all of the facts which designers, draftsmen and others need in all classes of machine design.

The contents of the book may be classified, roughly, in three parts: First, the data in common use, available by itself but essential to work of this character; second, the best of the contributions to the technical press and proceedings of technical societies which are not easily accessible, and, third, the contributions of specialists who have given freely of their data to make this work as broadly useful as possible. A better idea, however, of the broad scope and remarkable completeness of the book can be gained from the list of subjects discussed. The first half of the book includes chapters on mechanical principles of design, plain or sliding bearings, ball and roller bearings, shafts and keys, belts and pulleys, fly-wheels, cone pulleys and back gears, spur gears, bevel gears, friction gears, worm gears, helical gears, planetary gears, ropes, chains, brakes, friction clutches, cams, springs, bolts, nuts and screws, wire and metal gages, hydraulics and hydraulic machinery, pipe and pipe joints, minor machine parts, press and running fits and balancing machine parts. Following this are notes on miscellaneous mechanisms and the performance and power requirements of tools, and then several chapters are devoted to the materials commonly used in machine construction, taking up in turn cast iron, steel and the various alloys. A separate chapter is devoted to the construction and proportions of steam boilers, while there are similar chapters on the steam engine, the gas engine and compressed air. Finally there are chapters on mechanics and the strength of machine parts and over fifty pages of useful tables.

To present such an enormous mass of useful data in suitable form for immediate reference within the limits of a book of this size, and still treat each subject adequately, is a task of no small magnitude, and the author has shown excellent judgment in the form of presentation as well as in the skillful use of illustrations, charts, diagrams and tables. The size of page, style of type and form of charts and tabulated matter all aid in making a well balanced, useful reference book for machine designers.

NOVEL DRY DOCK FOR TORPEDO BOATS.—The dock construction firm von Klitzing, of Hamburg, Germany, has received a contract from the German navy for the construction of a novel type of dry dock for torpedo boats. The dock will consist of a lifting section and several floating pontoons. When docking a torpedo boat a pontoon will be sunk within the dock to receive the boat, and then both the dock and pontoon will be pumped out together. The suction valves of the pontoon will then be closed and the dock sunk sufficiently to enable the pontoon to be floated out with the torpedo boat on its deck. A second pontoon can then be placed in the dock and another boat docked in the same manner. The advantage of this arrangement over the ordinary floating dry dock is the possibility of docking any number of torpedo boats at the same time by using additional pontoons. As the pontoons carrying torpedo boats can be easily transported, and as they have a very shallow draft, they can be brought close to the shore, thus facilitating quick repairs on the craft that are being docked.

PANAMA PACIFIC LINE.—The International Mercantile Marine Company, comprising the American, Atlantic Transport, Red Star, White Star and White Star-Dominion lines, has announced that early in 1915 a new passenger and freight service, to be known as the Panama Pacific Line, will be established, with sailings every three weeks, between New York and San Francisco through the Panama Canal. The steamers selected for this route are the *Finland* and *Kronland*, each of 22,000 tons displacement, formerly employed in Red Star Line's New York-Dover-Antwerp service.

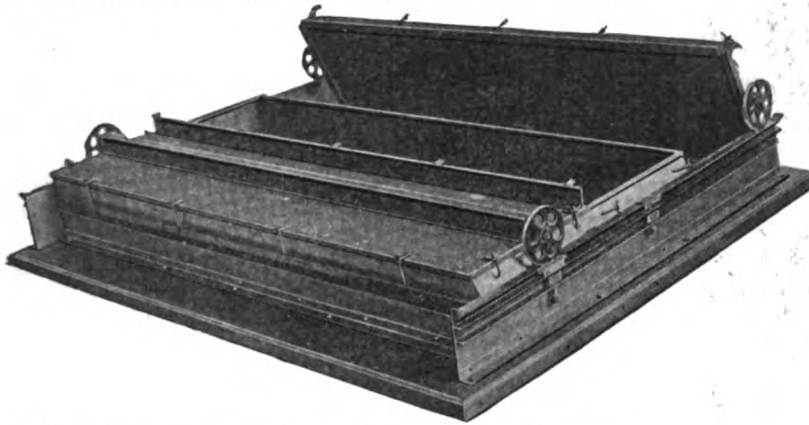
ENGINEERING SPECIALTIES

Underwriter Pivot-Balance Hatch Cover

Captain Arthur N. McGray, of 119 West Seventy-first street, New York City, an active steamship master of wide experience, has invented an improved type of steel hatch cover which he is now arranging to place on the market.

Steel covers, hinged to the coamings, which can be removed only by means of heavy tackle, uplifting them into a vertical position, have long been in effective use; but the disadvantage of always requiring steam power to handle them, and the first cost and upkeep of the equipment necessary to their safe handling, have militated to their universal adoption. Besides, in ships engaged in dry-cargo trades, particularly in climates where frequent heavy and passing showers are encountered, any type of cover that demands more than a few minutes of time, the expenditure of any considerable power, or experienced care in handling, is looked upon with disfavor. In the new pivot-balance type of cover illustrated, it is claimed that all these objectionable features have been minimized or altogether eliminated.

Structurally, all its elements are designed to standard dimension products of the rolling mills, thereby reducing the



problem of construction to merely punching and riveting. Channel or bulb-iron "stiffeners" are riveted to the upper side of the cover-plate, around the under edge of which runs a deep channel bar (into which the water-tightening material is fixed), which fits closely over the half-round of the coamings, so that not only do the toggle bolts draw the cover plate down watertight, but the channel bar acts as an efficient stiffener to the side and end coamings. Thus, the entire space enclosed by the coamings is free and clear of all obstruction for cargo stowing purposes. Along the sides of the coamings, fore and aft, runs a light trackway of steel. Midway of the fore-and-aft length of each cover-section there is fitted to its upper side an ordinary axle, carrying a small wheel, which rests upon the trackway mentioned. When any cover-section is at its normal position the wheel rests on a "filler-piece" which is cut out of the trackway. This filler-piece is supported on a jack-screw block which is coarse threaded, to move the cover up or down very quickly. Thus, the cover sections are lifted off the coamings and trundled into the space provided for them, where they may be pivoted into a vertical or semi-vertical position, all depending upon the manner in which the deck plan of the ship has been designed for their accommodation.

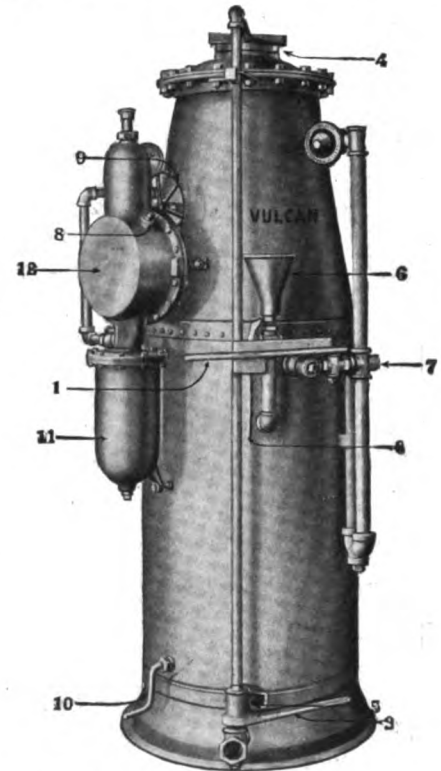
To replace a cover-section, pivot-balance it into nearly a horizontal position (one man at each side) and trundle it back until its wheels rest upon their own particular filler-pieces; then lower away the ratchet screw till the weight of the cover rests upon the coamings; throw up the toggle bolts into the forks on the cover and set up their butterfly nuts. When this is done, the battening down is completed.

Where a plurality of cover sections are required, as in the case of extremely large hatchways, the sections may be loosely coupled together by iron bars and all run off in one operation. In the Great Lakes' type of bulk carrier it is estimated that six men could put on and batten down, ready for any kind of weather, 32 hatches in a half hour, or take them off in 20 minutes.

Complete and simple provision has also been made to accommodate the principles of this design to all classes of craft—form barges and canal boats to "tramp" or ocean liners.

A Unique Acetylene Generator

An acetylene generator that employs a new and ingenious principle to meet the very exacting demands of an autogenous welding outfit is being marketed by the Vulcan Process Company, of Minneapolis, Minn., and Cincinnati, O. In this ma-



chine the feed mechanism drops an automatically measured quantity of $1\frac{1}{4}$ by $\frac{3}{8}$ -inch carbide into the water, varying the quantity to suit the demands made on the gas supply, producing a clean, cool gas at unvarying pressure. The motor that drives this automatic feed utilizes the buoyancy of the gas passing from the generator to the torch; thus the feed is increased as the gas consumption increases, or lessened when the gas consumption lessens, or the feed is automatically stopped and started when the torch is turned off or on. If the pressure for any reason should tend to rise above normal, the gas is conducted through a by-pass, rendering the feed inoperative until sufficient gas is used to lower the pressure. Possible accidents due to puncture are eliminated, it is claimed, by locating the feed motor in the pipe between the generator and the torch, and using the passing gas as motive power, instead of following the usual method of utilizing the gas pressure in the machine.

Being designed to use $1\frac{1}{4}$ by $\frac{3}{8}$ -inch carbide, it is claimed the machine will deliver 15 percent more acetylene than if the same quantity of screenings of carbide were used, and better gas results from the carbide falling deep into the water before complete decomposition ensues, securing cooler genera-

tion than is possible with screenings, which have a tendency to decompose near the surface, causing generation under high temperature and failure to give the gas the benefit of rising through a considerable volume of water, whereby it is washed and cooled. The carbide chamber and feed mechanism are removable, thus opening the machine for complete inspection. The entire generator is protected against careless manipulation by a locking device, which prevents removing the cap for refilling or opening any valves without following a definite safe routine.

Efficiency of Welded Joints as Applied to the Manufacture of Warehouse Trucks

A striking example of the increased efficiency and generally decreased cost of manufacture by welding sheet metal articles compared with drilling and riveting is shown on the steel trucks made by the Standard Motor Truck Company, of Chicago.

These warehouse trucks were formerly made by drilling and riveting all of the joints. Welding by the Oxweld process



was proposed and a test truck made by this method. The results were so conclusively in favor of this construction that it was immediately adopted and the riveting process abandoned.

Welding not only produced a one-piece truck in which the welded joints proved far stronger than the riveted joints formerly used, but it has increased the output per man about 20 percent with a saving of over 30 percent of the previous cost of manufacture per truck.

Tests of both the welded and riveted joints used in the manufacture of these trucks were made by the Bureau of Inspection, Tests and Construction of Robert W. Hunt & Company, engineers, Chicago. Three specimens were tested; the first consisted of a stringer tube and a cross tube welded together by the Oxweld process. Under a maximum load of 25,460 pounds the cross tube broke from the stringer tube at the weld. The second specimen consisted of a cross tube riveted to a stringer tube by two $\frac{1}{4}$ -inch rivets in double shear, one in the stringer and one in the cross tube through a plug and bushing inserted in the cross tube. Under a maxi-

mum load of 4,740 pounds the rivet in the stringer tube sheared off. The third specimen consisted of a cross tube joined to a stringer tube by a $\frac{1}{4}$ -inch rivet in double shear in the cross tube and a $\frac{1}{4}$ -inch riveted reduction of a plug inserted through the stringer tube, the head of the plug being in tension. Under a maximum load of 5,800 pounds the rivet in the cross tube sheared off and the riveted connection in the stringer tube pulled partially through the stringer tube.

Personal

HARRY WISSNER has been appointed assistant to Mr. Miller on the steamer *Olcott*.

NELSON A. CARTIER has been engaged as chief engineer on the steamer *H. A. Bagster*, of Norfolk, Va.

CAPTAIN O. G. HAINES has been appointed local inspector of steam vessels at Boston to succeed J. F. Blaine, resigned.

CAPTAIN WILLIAM McCANN and Engineer Alfred Pardee were recently placed in charge of the tug *W. B. McCullough*, Albany, N. Y.

THOMAS E. BEASLEY has accepted the position of assistant sales manager of the American Engine & Electric Company, Bound Brook, N. J.

DUDLEY S. JOHNSON succeeds the late Sam Mayer as manager of the Chicago branch of the Joseph Dixon Crucible Company, Jersey City, N. J.

MERRILL MARDON has been appointed to fill the position of chief engineer of the steamer *Angler* of the Potomac River, in place of B. O'Donnell, resigned.

ROBERT BURKE has been appointed chief engineer of the steamer *Alice M. Gill* of the Homegardener Sand Company. Mr. Burke will be assisted by Henry Burkley.

WILLIAM TIBBY, formerly chief engineer of the steamer *A. E. Shores, Jr.*, has been appointed chief engineer of the steamer *Protection* of the Henderson Sand Company.

DANIEL GRIEVES has been appointed to succeed John Hilton as chief engineer of the steamer *C. H. Little*, owned by the United Fuel and Supply Company, of Detroit, Michigan.

WILLIAM EVANS is chief engineer and Captain A. E. Bailey commander of the steamer *Austrana*, owned by the Furness Company. Her home port is West Hartlepool, England.

H. E. BLOM, assistant inspector of hulls at Baltimore, has been transferred to San Juan, P. R., as local inspector, to succeed W. K. Martin, who has been transferred to Boston.

H. W. KEITH has recently been promoted from the grade of instructor to the grade of assistant professor in naval architecture at the Massachusetts Institute of Technology, Boston, Mass.

WILLIAM GARDINER, formerly of the New England Navigation Company, has been engaged as chief engineer on the steamer *Isis* of the Bridgeport Towing Company, Bridgeport, Conn.

JOHN R. LA FOE, who was engineer on the steamer *John H. Hogan*, has accepted the position of chief engineer on the steamer *City Point* of the Seal-Ship Company, New Haven, Conn.

PROF. I. B. ABELL has been appointed professor of naval architecture at the Liverpool University, succeeding his brother, Prof. W. S. Abell, who is now chief surveyor to Lloyd's Register of Shipping.

JOHN SEEMAN, formerly chief engineer of the steamer *Albert V. Gowen*, has been appointed chief engineer of the new steel

steamer *Kelley's Island* of the Kelley's Island Lime and Transport Company.

CAPTAIN JAMES WOOD is in charge and Albert Hayck is chief engineer of the small side-wheel passenger vessel *Victor* of the Albany and Troy Steamboat Company, which went into commission July 5.

CAPTAIN CHARLES PICKETT was transferred from the tug *Mabel* to the tug *R. G. Davis*, and Lemuel Herrington was transferred to the same tug from the *George M. Southwick*, to act as chief engineer.

CAPTAIN GEORGE E. COOLEY was at the wheel and Elsworth B. Mealy had charge of the engines of the tug *Charles C. Wing*, while the vessel made its trial trip on July 18 last, after having been partly rebuilt.

JOHN J. HUGHES has been appointed to succeed David A. Black, resigned, as chief engineer of the steamer *City of Toledo* of the White Star Line, plying between Detroit and and Port Huron, Michigan.

CARL BECKER, formerly chief engineer of the steamer *Edward Recor* of the Kelley's Island Lime and Transport Company, has been appointed chief engineer of the steamer *Alva S. Chisholm* of the same company.

AUGUST SCHENK, formerly chief engineer of the steamer *Alice M. Gill* of the Homegardener Sand Company, has accepted the position of chief engineer of the steamer *Edward Recor* of the Kelley's Island Lime and Transport Company.

CAPTAIN JAMES H. DOLAN, who was formerly superintendent on barge canal work, is now in charge of the twin-screw passenger steamer *Julia Safford*, which went into commission on July 5 last. John J. Kenny is chief engineer of the *Julia Safford*.

EDWARD CONKLIN, of Rondout, has been chief engineer of the New York towing steamer *William H. Bavier* since it was built. This vessel recently arrived at Albany commanded by Captain Herbert Du Mont, of Rensselaer, with two oil tanks for the Texas Oil Company.

LOUIS MILLER, formerly chief engineer of the steamer *Alva S. Chisholm* of the Kelly's Island Lime and Transport Company, has accepted the position of chief engineer of the steamer *Olcott*, which has been chartered to ply between Oswego, N. Y., and the Thousand Islands.

FREDERICK O. BALL resigned his position as general manager of the American Engine and Electric Company, of Bound Brook, N. J., on July 1, to engage in the manufacture and sale of carburetors with his father, Frank H. Ball, under the firm name of Ball & Ball, Detroit, Mich.

CAPTAIN EDWARD SCOTT is in command and Henry Matte chief engineer, with Archie Winkler first assistant, on the towing steamer *Empire* of the Albany Towing Company. The *Empire* has been chartered by the Great Lakes Dredging & Dock Company for work on the Hudson River.

HARRY J. MARKS, manager of the New York office of the American Engine & Electric Company, Bound Brook, N. J., has been promoted to the position of sales manager of the company. Mr. Marks is a member of the American Society of Mechanical Engineers and other engineering societies.

JOHN SHERRATT, general superintendent of the White Star Line at Genoa, is about to retire after thirty-two years of active service, and has been presented with a silver salver by his employers. Mr. Sherratt has very ably acted in the capacity of general superintendent since he was appointed in 1904.

J. W. HAMILTON, director of Messrs. William Hamilton &

Company, Ltd., shipbuilders, Port-Glasgow, was elected president of the Shipbuilding Employers' Federation at its annual meeting held recently in Edinburgh. Mr. Hamilton succeeds Mr. Herbert B. Rowell, shipbuilding director of Messrs. R. & W. Hawthorn, Leslie & Company, Ltd., Hebburn.

HUGH M. MACMILLAN has been appointed manager of the shipbuilding department of the Fairfield Shipbuilding & Engineering Company, Ltd., of Govan, Glasgow, with a seat on the board of directors. Mr. Macmillan succeeds Mr. Alexander W. Sampson, who has resigned after fourteen years' service as manager of the Govan firm. The new manager was formerly general manager of the shipbuilding yards of Messrs. Workman, Clark & Company, Ltd., of Belfast.

Obituary

CHARLES F. MAGOUN died on July 5 last. Mr. Magoun was treasurer of the Hyde Windlass Company, of Bath, Maine.

JOSEPH G. ROBERTSON died recently from acute pneumonia at his home, 6 Claremont road, Forest Gate. He was the eldest son of Mr. A. W. Robertson, of Messrs. A. W. Robertson & Company, of the Lea Shipbuilding Yard, Canning Town.

ENGINEER REAR-ADMIRAL WISHART died recently in his sixty-second year. In the many important posts that he occupied in His Majesty's navy, his personal characteristics and administrative qualities won for him the respect and admiration of all those with whom he came in contact.

CHARLES HENRY LIBBEY, president of the Graphite Lubricating Company of New Jersey, died recently at Bound Brook, N. J., aged 62. Mr. Libby was for thirty years connected with the H. B. Claflin Company, New York, and resigned to accept the presidency of the Graphite Lubricating Company twelve years ago.

MATTHEW CLOVER, a well-known shipbuilder, recently of the firm of Messrs. Clover, Clayton & Company, of Birkenhead, died at his residence at Willaston, Cheshire, in his sixty-seventh year. Mr. Clover had been active in shipbuilding until he resigned about three years ago from the firm with which he had been associated for forty-seven years.

FRANCIS JAMES TREWENT, recently of the firm of J. F. Trewent and Proctor, Ltd., died on May 17 last at his residence at Hampstead as a result of heart seizure. As an engineer Mr. Trewent was well known, not only to naval architects, but also to a large number of people interested in the mercantile marine. At the age of sixteen he entered the Royal School of Naval Architecture, at South Kensington, where he had the companionship of fellow students who later attained the highest distinction in that branch of engineering. Mr. Trewent was a member of the Institution of Civil Engineers, and other technical societies.

ALEXANDER DUNBAR, who had been appointed chief engineer of the new Cunard liner *Aquitania*, died suddenly on April 4 at Whiteinch, Glasgow, from a severe attack of pneumonia. He had not completely recovered when, anxious to be out during the completion of the *Aquitania*, he resumed his overseeing duties on that vessel. Mr. Dunbar had been in the the Cunard service since May, 1883, and previous to his appointment to the *Aquitania* he was chief engineer on the *Lusitania* from February, 1911, and took her on her maiden voyage under the direction of Mr. John Currie, consulting engineer to the Cunard Line, under whom Mr. Dunbar had also served on the *Mauretania*.

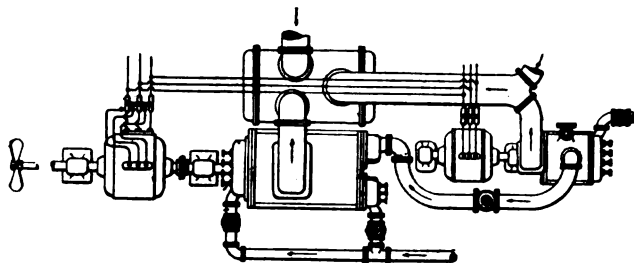
SELECTED MARINE PATENTS

The publication in this column of a patent specification does not necessarily imply editorial commendation.

American patents compiled by Delbert H. Decker, Esq., registered patent attorney, Millerton, N. Y.

1,095,461. SHIP PROPULSION. WILLIAM L. R. EMMET, OF SCHENECTADY, N. Y., ASSIGNOR TO GENERAL ELECTRIC COMPANY, A CORPORATION OF NEW YORK.

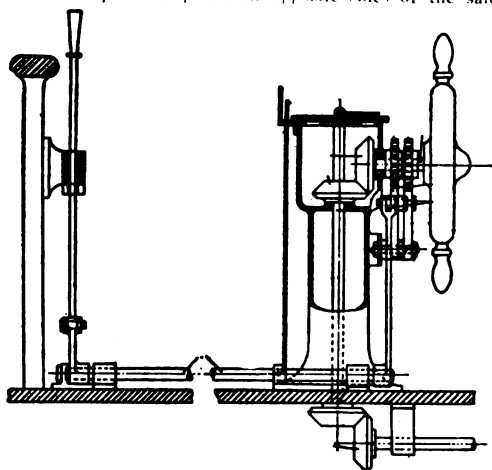
Claim 18.—In a system of ship propulsion, the combination of a turbine constructed and arranged to operate condensing and non-condensing, means for cutting a portion of the turbine buckets out of service for non-condensing operation, an electric generator driven by the turbine, a propeller and its shaft, an electric motor mounted on the shaft which receives its current from the generator, the generator and motor driving



the ship at certain speeds, a low pressure turbine whose rotor is connected to the propeller shaft, means for admitting steam from the generator-turbine to the low-pressure turbine when it is desired to propel the ship at higher speeds, condensing means for the turbines, and means for cutting the electrical apparatus out of service and admitting steam to the low-pressure turbine when it is desired to maneuver the ship. Twenty claims.

1,100,420. SHIP'S STEERING GEAR. JAMES RICHARD CLAY, OF BOOTLE, ENGLAND.

Claim 1.—Steering gear of a ship comprising in combination a steering gear; a spindle fitted with a hand wheel for operating the gear, and with a pair of ratchet wheels having teeth oppositely formed; a pair of connective pawls disposed on opposite sides of the said spindle



and each engaging with one of the ratchet wheels; lever mechanism connected with the pawls; and means connected with the lever mechanism for operating the same, and for moving the pawls to and from the ratchet wheels, substantially as described. Two claims.

1,099,971. GUIDE BLOCK FOR TORPEDOES. WILLIAM J. DOOLAN, OF WASHINGTON, DISTRICT OF COLUMBIA.

Claim 1.—The combination of a torpedo spoon, a torpedo disposed in said spoon and slidably interlocking therewith, and means interposed between the interlocking portions of the torpedo and spoon for absorbing friction incident to the discharge of the torpedo irrespective of the direction of transverse strains exerted upon the torpedo during such discharge. Six claims.

British patents compiled by G. F. Redfern & Company, chartered patent agents and engineers, 15 South street, Finsbury, E. C., and 10 Gray's Inn Place, W. C., London.

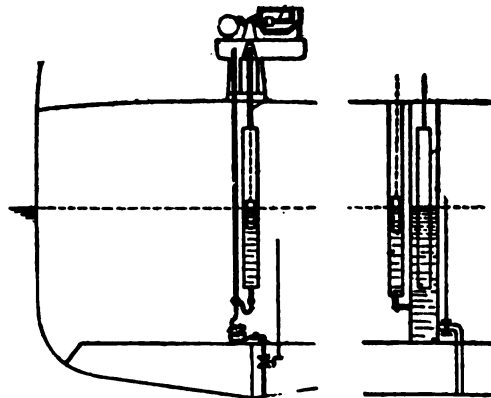
13,755. APPARATUS FOR DISCHARGING ASHES AND OTHER MATERIALS FROM SHIPS. WILLOUGHBY LAKE BAY, LAY, OF BLACKHEATH, ASSIGNOR TO J. STONE & CO. LTD., OF DEPTFORD.

Claim 1.—Apparatus for discharging refuse from ships below the waterline comprising in combination, a receptacle for the refuse, a discharge duct, a valvular device between said receptacle and duct and adapted to alternately receive successive charges of refuse and to deliver same to said duct, a discharge connection between the valvular device and the duct, a water pipe discharging below the water line of the ship in connection with said duct and adapted to carry water past

the delivery end of the discharge connection, a pump in connection with said water pipe adapted to produce a continuous stream of water therein, a compressed air supply pipe connected with the discharge connection in such a manner as to prevent water from rising into said valvular device, and means adapted to relieve the pressure from the valvular device prior to each opening of the valvular device to the refuse receptacle, substantially as set forth. Four claims.

26,903/1912. IMPROVED APPARATUS FOR SIMULTANEOUSLY WEIGHING THE CARGOES AND MEASURING THE DRAFT OF SHIPS. C. FINCK, OF HOHENZOLLERNRING 54, KIEL, GERMANY.

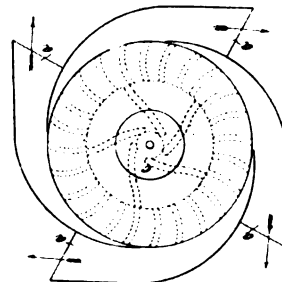
Comprises apparatus for measuring the loading of ships by means of a measuring member suspended away from the center of gravity of the ship from a balance, and forming, on a smaller scale, a reproduction of the ship's form, and capable of being raised or lowered through a distance, which depends on the slant of the ship and on the distance between the measuring vessel and the ship's center of gravity with the



existing condition of loading. A device for directly indicating the draft at several places in the ship's length, comprises essentially a pointer, actuated by a float, moving over scales adjusted by a water level, according to the trim of the ship. To determine the distance through which the measuring members must be raised or lowered when the ship is out of trim, a second device is employed consisting of a chart drum operatively connected to the water level to enable that distance to be read off.

27,642/1912. FANS FOR FORCING AIR INTO THE STOKEHOLDS OF TORPEDO BOAT DESTROYERS, ETC. W. H. ALLEN AND R. W. ALLEN, BOTH OF QUEEN'S ENGINEERING WORKS, BEDFORD.

Claim.—Relates to fans for forcing air into the stokeholds of torpedo boat destroyers and other small craft where the space is very limited, and consists in the combination with a fan of the type shown



situated at the top of a stokehold and wholly within it and sucking air through an inlet in the wall of the stokehold of a casing provided with one or more outlets delivering air at the top of the stokehold in a direction away from the deck and sides and the beams supporting them. One claim.

19,192/1912. APPARATUS CONTROLLING CLOSING OF BULKHEADS DOORS FROM A DISTANCE. W. G. GIBBONS, OF ROSEBANK IRON WORKS, EDINBURGH, AND J. BLAKE.

Claim.—This invention relates to bulkhead doors which close by gravity when released, the object being to provide improved and simple apparatus by which the clutches, pawls, etc., in the operating mechanism may be withdrawn from the ship's bridge. The apparatus comprises a disconnectable locking device in each door operating gear, means controlled by a spring and by a small hydraulic cylinder to operate the locking device, a large hydraulic cylinder at the transmitting station, valve or cock controlled pipe connections putting each of the small cylinders in connection with the large cylinder, means for operating the ram or piston of the large cylinder by power or by hand, or by both, an operating-fluid supply tank situated above the apparatus, a supply pipe connecting this tank with the large hydraulic cylinder, and means for closing the supply pipe on the inward movement, so that when hydraulic pressure is transmitted the locking device is withdrawn and the door is free to close. One claim.

1,109,1913. STEERING AND CONTROLLING STEAMSHIPS AND THE LIKE. S. WHITAKER AND S. W. COOPER, OF L. 49 ROYAL EXCHANGE, MANCHESTER.

Improved means for controlling the steering of steamships, etc., consists in the employment below the waterline of the vessel and at either the bow or stern or at both ends, of one or more pipes open at one end to the water at one side of the vessel and at the other end to the other side of the vessel for the exit of water, a rotary pump being arranged in each pipe for inducing a flow of water through each pipe from one side of the vessel and to discharge it at the other side.